RECOMMENDED PRACTICE
DNV-RP-D101

STRUCTURAL ANALYSIS OF PIPING SYSTEMS

OCTOBER 2008

DET NORSKE VERITAS

FOREWORD

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G) Asset Operation
H) Marine Operations
J) Wind Turbines
O) Subsea Systems

Amendments and Corrections

This document is valid until superseded by a new revision. Minor amendments and corrections will be published in a separate document normally updated twice per year (April and October).

For a complete listing of the changes, see the “Amendments and Corrections” document located at: http://webshop.dnv.com/global/, under category “Offshore Codes”.

The electronic web-versions of the DNV Offshore Codes will be regularly updated to include these amendments and corrections.

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1. General

1.1 Introduction

This Recommended Practice is based on, and intends to, show the best from European industrial practice for structural analysis of piping systems intended for the offshore sector. Typical applications are Oil & Gas Platforms, FPSOs, Drilling Units and Subsea installations. Subsea installations are installations such as templates, manifolds, riser-bases and subsea separation-and pump modules.

There is no piping design code that fully covers these topics, and hence Engineering Companies have developed a variety of internal design philosophies and procedures in order to meet the requirements to structural integrity, safety, economical and functional design of piping systems.

A number of references are given to below listed codes and standards from which equations for a large number of pipe-stress relevant calculations can be found.

1.2 Objective

The objective of this recommended practice is to describe "a best practice" for how structural analysis of piping systems can be performed in order to safeguard life, property and the environment. It should be useful for piping structural engineers organising and carrying out the piping design, and any 3rd party involved in the design verification, such as Class Societies, Notified Bodies etc. The proposed project documentation should provide the operator with essential design information and be useful during commissioning, maintenance, future modifications, and useful in order to solve operational problems, if and when they occur.

1.3 Relationship to other codes

This Recommended Practice is strongly related to the use of a large number of international design codes, standards, directives and regulations in order to succeed with a professional design.

1.4 References

The below listed codes, standards, recommended practices, specifications and software are considered to be the most important ones for analysis of piping systems and piping components to be installed in an offshore environment.

1.4.1 ASME codes and standard

ASME B16.5	Pipe Flanges and Flange Fittings
ASME B16.9	Factory Made Wrought Steel Butt welding Fittings
ASME B16.20	Metallic Gaskets for Pipe Flanges-Ring Joint, Spiral Wound, and Jacketed
ASME B16.28	Non-metallic Flat Gaskets for Pipe Flanges
ASME B16.47	Large Diameter Steel Flanges
ASME B16.49	Factory Made Wrought Steel Butt welding Induction Bends for Transportation and Distribution Systems
ASME B31.1	Power Piping
ASME B31.3	Process Piping
ASME B31.4	Pipeline Transportation Systems for Liquid Hydrocarbons and Other Liquids
ASME B31.8	Gas Transmission and Distribution Piping Systems
ASME B36.10M	Welded and Seamless Wrought Steel Pipe
ASME B36.19M	Stainless Steel Pipe
ASME BPVC Sect. II	Part D Material Properties
ASME BPVC Sect. III	Rules for Nuclear Facility Components, Division 1
ASME BPVC Sect. VIII	Rules for Construction of Pressure Vessels, Division 1 & 2

1.4.2 API Codes and Standards

API 6A	Specification for Wellhead and Christmas tree Equipment
API 6AF	Capabilities of API Flanges under Combination of Load
API RP 2A-WSD	Recommended Practice for Planning, Designing and Constructing Fixed Offshore Platforms - Working Stress Design
API RP 2FB	Recommended Practice for the Design of Offshore Facilities against Fire and Blast Loading
API RP 14E	Recommended Practice for Design and Installation of Offshore Production Platform Piping Systems
API RP 17A	Design and Operation of Subsea Production Systems (API equivalent to ISO 13628)
API RP 520	Sizing, Selection and Installation of Pressure-relieving Devices in Refineries
API Std. 610	Centrifugal Pumps for Petrochemical and Natural Gas Industries
API Std. 611	General-Purpose Steam Turbines for Petroleum, Chemical and Gas Industry Services
API Std. 616	Gas Turbines for Petroleum, Chemical, and Gas Industry Services
API Std. 617	Axial and Centrifugal Compressors and Expander-compressors for Petroleum, Chemical and Gas Industry Services

1.4.3 Other Standards and Bulletins

AISC ASD	American Institute of Steel Construction, Allowable Stress Design
AISC LRFD	American Institute of Steel Construction, Load and Resistance Factor Design
EJMA	The EJMA Standards for design, installation and use of expansion bellows
NEMA SM23	Steam Turbines for Mechanical Drive Service
WRC 107	WRC Bulletin No. 107. Local Stresses in Spherical & Cylindrical Shells due to External Loadings
WRC 297	WRC Bulletin No.297. Local Stresses in Cylindrical Shells Due to External Loadings on Nozzles. Supplement to WRC Bulletin No. 107.
WRC 449	Guidelines for the Design and Installation of Pump Piping Systems

1.4.4 DNV Offshore Standards

DNV-OS-F101	Submarine pipeline systems

1.4.5 DNV Recommended Practices

DNV-RP-C203	Fatigue strength analysis of offshore steel structures
DNV-RP-C205	Environmental Conditions and Environmental Loads
DNV-RP-F112	Design of Duplex Stainless Steel Subsea Equipment Exposed to Cathodic Protection

1.4.6 European Codes and Standards

BSI BS 7159	Code of Practice for Design and Construction of Glass Reinforced Plastics (GRP) piping systems for Individual Plants or Sites.
EN-1591.1	Flanges and their joints. Design rules for gasketed circular flange connections. Calculation method.

EN 1993	Euro code 3, Design of Steel Structures
EN 13480	Industrial Metallic Piping
EN 13445	Unfired Pressure Vessels
EN-ISO-13628	Design and operation of Subsea Production Systems
EN-ISO 13703	Petroleum and Natural Gas Industries-Design and installation of Piping Systems on Offshore production Platforms
EN-ISO- 14692	Petroleum and Natural Gas Industries-Glass Reinforced Plastic Piping (GRP)
PD 5500	Specification for Unfired Fusion Welded Pressure Vessels
PED	Pressure Equipment Directive (97/23/EC)

1.4.7 NORSOK Standards

NORSOK L-001	Piping and Valves
NORSOK L-002	Piping Design, Layout and Stress Analysis
NORSOK L-005	Compact Flanged Connections
NORSOK M-001	Material Selection
NORSOK M-630	Material Data Sheets for Piping
NORSOK N-001	Structural Design
NORSOK R-001	Mechanical Equipment

1.4.8 Other historical important piping publications

M.W. Kellogg	Design of Piping Systems
Roark's	Formulas for Stress and Strain
CASTI	Practical Guide to ASME B31.3
MDT	Guidelines for the Avoidance of Pipework Fatigue
CMR	Explosion Handbook

1.4.9 Pipe Stress Software

Auto Pipe	Bentley Systems, Exton, Pennsylvania, USA
CAE Pipe	SST Systems Inc. San Jose, California, USA
Caesar II	Coade Inc. Houston, Texas, USA.
PipePak	Algor Inc. Pittsburg, Pennsylvania, USA
Triflex	Piping Solutions Inc. Houston, Texas, USA.

1.5 Definitions

1.5.1 Verbal forms

Shall: Indicates requirements strictly to be followed in order to conform to this RP and from which no deviation is permitted.

Should: Indicates that among several possibilities, one is recommended as particularly suitable, without mentioning or excluding others, or that a certain course of action is preferred but not necessarily required. Other possibilities may be applied subject to agreement.

May: Verbal form used to indicate a course of action permissible within the limits of the RP.

Recommend: Indicates the preferred method. Other suitable alternatives may be permitted subject to agreement.

1.6 Abbreviations and Symbols

1.6.1 Abbreviations

AISC	American Institute of Steel Construction
ALS	Accidental Limit State
ASD	Allowable Stress Design
ASME	American Society of Mechanical Engineers
ASTM	American Society for Testing and Materials
API	American Petroleum Institute
BPVC	Boiler and Pressure Vessel Code (ASME Code)
CP	Cathodic Protection
CTOD	Crack Tip Opening Displacement
DAF	Dynamic Amplification Factor (in the context of this RP the same as DLF)
DFF	Design Factor Fatigue
DLF	Dynamic Load Factor
DFO	Documentation For Operations
DN	Diametre Nominale. (Nominal outer diameter of a pipe [mm])
DNV	Det Norske Veritas
DNV OS	DNV Offshore Standard
DNV OSS	DNV Offshore Standard Specification
DRWG	Drawing
EJMA	Expansion Joint Manufacturers Association
ESD	Emergency Shut Down
ESDV	Emergency Shut Down Valve
FABIG	Fire and Blast Information Group
FE	Finite Element
FEA	Finite Element Analysis
FEM	Finite Element Method
FLACS	FLame ACceleration Simulator (3D software used to simulate explosions)
FLS	Fatigue Limit State
FPSO	Floating Production Storage and Offloading Vessel
GRP	Glass Reinforced Plastics
HAZ	Heat Affected Zone
HD	Hold Down (vertical restraint, resists pipe lift-off)
HE	Hydrogen Embrittlement
HISC	Hydrogen Induced Stress Cracking
HSE	Health Safety and Environment (discipline)
HSE	Health & Safety Executive (body located in United Kingdom)
ISO	International Standards Organisation (also used for a piping standardised drawing)
LRFD	Load and Resistance Factor Design
LG	Line Guide (one or two directional restraint, perpendicular to pipe axis)
LS	Line Stop (restraint used to resist the axial movement of a pipe)
MOU	Mobile Offshore Unit
NEMA	National Electrical Manufacturers Association (US)
NDT	Non-Destructive Testing
NORSOK	Norwegian abbreviation. Norwegian Sector Offshore Standards.
NPS	Nominal Pipe Size
PED	Pressure Equipment Directive (97/23/EC)
PFP	Passive Fire Protection
PN	Pressure Nominale (Pressure rating class for flanges and valves. [bar])
PS	Pipe Support
PWHT	Post Weld Heat Treatment
ROV	Remotely Operated Vehicle (a subsea robot operated from a surface vessel, etc.)
RP	Recommended Practice
RS	Rest Support (vertical restraint that resist vertical movements caused by gravity)
SCF	Stress Concentration Factor
SDOF	Single Degree of Freedom
SIF	Stress Intensification Factor (in ASME context closer to SCF than the definition used in fracture mechanics)
SMYS	Specified Minimum Yield Stress
SMTS	Specified Minimum Tensile Strength
ULS	Ultimate Limited State
WRC	Welding Research Council (US)

1.6.2 Symbols

Symbols are described in the relevant sections where they apply. This RP gives references to sections in above listed codes and standards for a large number of different pipe-stress structural calculations. Relevant symbols are defined within these codes and standards.

2. Structural Analysis of Piping Systems

2.1 Introduction

2.1.1 General

Structural analyses of piping systems is within the piping discipline commonly referred to as pipe stress analysis or just stress analysis.

To validate the structural integrity of piping systems the piping structural engineers (pipe stress engineers) performs different types of analysis dependent on the criticality, design code, international-and national regulations, load-cases and any specially request from customers such as optimisation with regard to total assembly- weight and volume.

2.1.2 Scope

The scope of this section is to give a brief and general overview of typical analysis carried out by pipe stress engineers, analysis tools and piping design codes. The other sections and appendices of this RP are more specific for the pipe stress engineer and any part performing 3rd party verification and may hence not be of interest to others.

2.2 Pipe stress and flexibility analysis

2.2.1 General

Some piping systems are more critical and more difficult to design than others with regard to temperature variations, vibrations, fatigue and connection to sensitive equipment such as turbines and compressors. The forces and moments that a poorly designed piping system can add to the pipe support structure and connected equipment can be enormous and detrimental. Flanges may separate, hydrocarbon gases escape and mixed with air detonate if there is an ignition source. Flange leakage of condensate or diesel dripping onto a hot surface may start a fire, and for other piping systems people may be exposed to toxic fluids and gases such as hydrogen sulphide and radiation. Cryogenic leakage from piping on LNG FPSOs is another hazard.

When the piping industry started, there was a well known understanding of how to design piping systems with regard to the internal pressure and assembly weight, however, piping cracked and it was necessary to get a deeper understanding of why. The main reason why piping exposed to large temperature cracked was poor welds and how the layout of the piping was arranged; if it allowed for snaking or buckling or free expansion and contraction due to large temperature variations or imposed displacements from other sources, it had a higher likelihood of survival. For very high temperatures material creep and aging was another phenomenon that was not known then but now has to be considered. The latter is however not very relevant for topside and subsea piping as the temperatures do not get that high. Exceptions may be flare tip piping and gas turbine exhaust piping.

2.2.2 Stress categorisation

2.2.2.1 Primary stresses

Primary stresses are those developed by the imposed loading and are necessary to satisfy the equilibrium between external and internal forces and moments of the piping system. Typical loads are dead weight and internal pressure. Sustained stresses are primary stresses. Primary stresses are NOT self-limiting.

2.2.2.2 Secondary stresses

Secondary stresses are those developed by constraining the free displacement of piping subjected to thermal loads or imposed displacements from movements of anchor points etc. Hence, thermal-and displacement stresses are in the secondary stress category. Secondary stresses are self-limiting.

2.2.2.3 Peak stresses

Unlike loading conditions of secondary stress which causes distortion, peak stresses cause no significant distortion in piping. Peak stresses are the highest stresses in the region under consideration and should always be taken into consideration in fatigue and fracture mechanic calculations.

2.2.3 Flexibility analysis

Flexibility analysis is performed in order to investigate the effect from alternating bending moments caused by pipe temperature expansion/contraction and other imposed displacements from e.g. thermal expansion of pressurised equipment, tall swaying scrubbers, FPSO swivel stacks, bridge-piping between a floating production platform and a fixed wellhead platform, live load deck deflections, sag and hog effect on a FPSO etc.

In flexibility analysis the issue is to design the piping systems in such a way that parts of the piping it self act as a spring and release or reduce the internal bending stresses or longitudinal stresses that otherwise would have been detrimental for a straight pipe run between equipment. Flexibility is added in a system by changes in the run direction (offsets, bends and loops) or by use of expansion joints or flexible couplings of the slip joint, ball joint or bellow type. In addition more or less flexibility can be added by changing the spacing of pipe-supports and their function (e.g. removal of a guide close to a bend to add flexibility).

Another way to increase the flexibility is to change the existing piping material to a material with a higher yield-or tensile strength, or to a material quality that does not need additional corrosion and erosion allowance, and thereby obtains a reduction in the wall thickness which again gives more flexibility since the moment of inertia is reduced with a reduction of the pipe wall thickness.

In order to improve the flexibility it also helps to change to a material with lower Young's Modulus. When changing carbon steels used for e.g. firewater ring main on a FPSO subjected to alternating sag and hog moments (ship- or vessel bending moments caused by sea waves) with Titanium or GRP materials in order to avoid problems with internal corrosion and maintenance, additional flexibility is gained as the Young's Modulus for Titanium and GRP are much lower than for carbon steels.

It should however be noted that materials with lower Young's Modulus will in general have lower allowable basic stress and that carbon steel systems replaced by thinner wall non corrosive materials could result in greater vibration.

Flexibility analysis should normally be extended to a simplified or formal fatigue analysis when there is more than one additional and essential cyclic load source, e.g. other sources than pure temperature cycles which are taken care of by the equation for displacement stress calculation in most piping design codes.

2.2.4 Stress analysis

Stress analysis of a piping system is closely linked to the flexibility analysis. Refer also to the sections below describing static and dynamic stress analysis. Stress analysis also includes the calculations of pipe wall thickness with regard to internal and external pressure, calculation of required reinforcements, and items such as maximum allowable vertical deflection (sag) in order to avoid pockets with fluid in a drained piping system prior to repair etc.

Time spent on pipe wall thickness calculations according to relevant design codes is only a fraction of the total time spent on pipe stress and flexibility analysis and normally not of any concern as this is covered by the project piping and valve specification and the referred piping class sheets.

Piping stress analysis is by international and national piping design codes more or less limited to linear- static and dynamic analysis. All piping design codes used today are also based on the traditional ASD (Allowable Stress Design) methodology, whereas some structural steel design codes and riser-and pipeline codes also include the LRFD (Load Resistance Factor Design) methodology.

Most of the pipe stress analysis carried out in projects is global pipe stress analysis of piping systems by using FEA software that is based on the beam element theory in combination with stress intensity-and stress concentration factors. Time spent on local design checks by hand calculations or FE analysis with solid-or shell elements is normally less than 5% of the total time spent on global pipe stress and flexibility analysis. Local design checks may include analysis of non-standard branch connections, pressure vessel nozzle-to shell analysis, additional pipe-wall membrane stresses caused by local interaction from pipe supports, special flanges, high frequency (acoustic) fatigue calculations etc. Below is a short description of typical pipe stress analysis techniques that are available to pipe stress engineers by use of modern FEA software specially made for global analysis of piping systems. These are: Static analysis, quasi-static analysis and dynamic analysis.

2.2.5 Static Analysis

Static analysis is the analysis carried out in order to find the sustained (primary) stresses, displacement (secondary) stresses, pipe support loads and equipment loading due to loads caused by the internal static pressure, deadweight of the pipe (including content, insulation, snow and ice accumulation, valves, etc.) and other sustained and displacement loads. Static analysis is considered mandatory for all piping systems requiring a comprehensive analysis

2.2.6 Quasi Static Analysis

Loads with a dynamic nature such as earthquake, blast winds from an explosion, water-hammer, slugs, pressure-surge and loads from pressure relieving devices such as PSV and rupture discs are commonly analysed by use of static models in combination with a load magnifier to simulate the maximum load response. The load magnifier is commonly referred to as the DLF (Dynamic Load Factor) or DAF (Dynamic Amplification Factor). Dynamic Load Factors that are selected from figures or charts depending on the load duration and natural frequency of the piping may be below 1.0, but should be chosen as high as 1.5-2.0 when the natural frequencies and load duration time are unknown, or otherwise when DLF figures for the actual component/or system being analysed are not established.

2.2.7 Dynamic Analysis

Dynamic analysis of piping systems consists of:

— Modal Analysis
— Harmonic Analysis
— Response Spectrum Analysis
— Time History Analysis.

2.2.7.1 Modal Analysis

Modal Analysis is carried out in order to find the piping natural frequencies and the associated mode shapes. A piping system consists of elastic components (pipes, bends, tees, spring supports etc.) and uneven distributed masses (varying pipe sizes and fittings, valves, flanges and other rigid components). Once displaced from static equilibrium, the system will oscillate at a combination of the mode shapes, each vibrating at the associated frequency.

Finding the piping systems natural frequencies are essential in order to determine the size of Dynamic Load Factors (DLF) and to determine the correct pipe-support spacing in order to avoid detrimental vibrations caused by internal flow, pressure transients, and vortex shedding oscillations from wind or sea currents passing over the piping.

Modal analysis of a static model is usually not time consuming and should therefore be carried out to determine the lowest natural frequency of the system. A typical system supported in accordance with a good pipe support standard should result in a lowest natural frequency not less than 4 to 5 Hz.

It is necessary to carry out Modal Analysis prior to other dynamic analysis such as Harmonic Analysis, Response Spectrum Analysis and Time History Analysis as these all use the piping natural frequencies obtained through the Modal Analysis. When performing modal analysis it is very important how the piping calculation model is build up. One can get very dissimilar results depending on the model details, and generally it is required to add more intermediate data points to obtain correct distribution of the masses.

2.2.7.2 Harmonic Analysis

Harmonic analysis determines the steady-state response of a linear structure to loads that vary sinusoidal with time such as “slugs” from piston driven pumps. These loads are modelled as displacements (or concentrated forces) at one or more points in the system. If the system will see multiple loads, the stress engineer will use phase angels to differentiate the loads from each other. Through the use of Harmonic Analysis on relevant piping systems, the stress engineer may sort out the maximum dynamic loads, stresses and deflections (amplitudes) that the system will see.

2.2.7.3 Response Spectrum Analysis

Response spectrum analysis may be used to account for exceptional loads such as earthquake. Prior to the response spectrum analysis a modal analysis has to be performed in order to obtain the natural frequencies for the individual modes of vibration. The maximum responses in each mode can be obtained using the response spectrum. The individual maximum modal responses are combined to obtain an estimate of the maximum piping system response.

Response spectrum analysis will often result in lower loads and hence a design that requires less steelwork than obtained from quasi-static analysis. This is because piping engineers in quasi-static analysis will use the maximum accelerations from the project “Design Basis” for all piping systems analysed rather than the actual value that can be extracted from the acceleration /frequency curve (response spectra curve) based on the systems natural frequencies.

2.2.7.4 Time History Analysis

Time History Analysis is used when the stress engineer wants to study the dynamic impact from time-dependent loads such as firing of a pressure safety valve, fast closing of an ESD valve or an uncontrolled start-up or break down of a pump. The latter phenomena are also referred to as fluid hammer and surge.

Earthquake and blast (hydrocarbon explosion) analysis can also be carried out by use of Time History Analysis. This may lead to much lower pipe-support reaction forces and stresses and deflections in the piping systems than obtained by quasi-static analysis. The reason is again that pipe stress engineers performing quasi-static analysis normally use conservative Dynamic Load Factors in the order 1.5-2.0.

2.3 Analysis tools

Piping codes will normally only allow for simplified hand calculations of a system’s flexibility if the system satisfies a certain equation with focus on the total pipe length and the straight line between two fixation points (equipment to equipment). No intermediate pipe supports are allowed in this method. This is however a rare and not common layout and is most applicable to small bore piping and tubing located within an engine etc. New piping systems that are not a duplicate of existing systems with a known history of successful operation are by most piping codes deemed to be analysed by extensive pipe stress calculations.

Today there is no other practical or economical way to perform extensive analysis and document that the analysis satisfies the piping code requirements than by use of dedicated and commonly used pipe stress software based on beam element theory or general purpose FEA programs with a piping code-check module.

There is no more than a handful of dedicated pipe stress software with a good reputation that is commonly in use by established pipe stress societies.

The intention of this RP is not to favour or disfavour any FE analysis tools used for piping design. It is however recommended to do a search on the WEB for “pipe stress software” and concentrate on software manufacturers that have a long history in development of mechanical- and pipe stress analysis tools, have a serious support department, WEB discussion forums, tutorial sections, organise courses etc. They should also be ISO certified and have a list of well known references.

Static and Dynamic Analysis of piping systems are described in detail in the user manuals supplied with such programmes and special courses are also organised by the manufacturer of pipe stress software.

In order to gain a deeper knowledge about the use of static and dynamic analysis tools than covered herein, it is strongly recommended to attend a beginner, intermediate or advanced pipe stress analysis course organised by the companies behind one of the 5 most sold pipe stress programmes. In addition it will be necessary to work close with experienced pipe stress engineers for some years.

2.4 Piping design codes

2.4.1 General

The piping industry is old and almost every country has its own design code for process piping. Hence it is essential that the pipe stress engineers are alert and take initiative to get updated on any national regulations regarding piping design if a power plant or offshore installation is going to be placed in foreign countries. There are special design codes for process metallic piping, subsea piping, fibre reinforced plastic piping and also special design codes for piping to be installed in nuclear power plants etc. This section will only concentrate on a short description of the most commonly used piping codes related to offshore installations.

2.4.2 ASME B31.3 Process Piping

The basic design code for engineers working with topside offshore projects is the *ASME B31.3 Process Piping Code.*

ASME B31.3 piping code has some basic requirements regarding the integrity of the piping but leaves all other aspects of the functional design of the system to the designer (in this context the pipe stress engineer). Due to the fact that the ASME B31.3 piping code does not address how a lot of mechanical pipe stress calculations should be performed, a number of textbooks and articles on how to interpret the ASME B31.3 code have been issued, even from the ASME B31.3 committee members, such as *CASTI “Practical Guide to ASME B31.3*”.

Many engineering companies also have their own “Pipe Stress Procedure” in order to describe how pipe stress analysis should be carried out in lack of guidance and requirements in the B31.3 piping code. (Example: How are piping systems to be designed with respect to exceptional- or accidental design loads from a hydrocarbon explosion? ASME B31.3 gives no guidelines).

2.4.2.1 ASME B31.3 requirement to education and practice

Unlike many other ASME and European design codes the ASME B31.3 piping code has some very strict requirements to the formal education and required training-or practice. This can be based on the fact that the code leaves much of the responsibility regarding the grade of necessity for additional structural integrity calculations and load cases to be judged by an experienced designer (pipe stress engineer).

One requirement to formal education and practice/training has been copied and pasted directly into this document from the latest revision (2006) of Chapter II, of the ASME B31.3 piping code:

> *(a)* Completion of an engineering degree, requiring four or more years of full-time study, plus a minimum of 5 years experience in the design of related pressure piping.

For pipe stress analysis this will in practice require a Bachelor or Master degree in Mechanical Engineering followed by minimum 5 years project-training in a piping department responsible for design and under supervision from senior-or principal pipe stress engineers.

According to ASME B31.3 a designer (pipe stress engineer) with a lower formal education than Bachelor- or Master Degree needs 10-15 years of practice and supervision in order to be responsible for the design (pipe stress analysis).

If the Engineering Company does not have personnel with the qualifications required according to chapter II of the code, then the piping design does not fulfil the requirements of the B31.3 piping code. Hence the “owner” (oil-company, etc.) or any 3rd party involved in the verification of the piping analysis should, if requested, be given a CV or statement from the manufacturer that proves the required formal qualifications of the personnel responsible for pipe stress analysis in the project.

2.4.3 EN-13480 Industrial Metallic Piping

The intention of the EN-13480 piping code was to harmonise a large number of European national piping codes in order to satisfy the essential safety requirements of the new European Pressure Equipment Directive, PED.

European countries have now made the EN-13480 design code to a National piping standard by adding national standards letter-codes in front of it, some examples are:

— BS-EN-13480British Piping Standard
— DIN-EN-13480German Piping Standard
— NS-EN-13480Norwegian Piping Standard.

EN 13480 is a more comprehensive piping design code with regard to pipe stress analysis than the ASME B31.3 Piping Code. Where the ASME B31.3 code leaves it to the designer to decide how to calculate some loads, the EN-13480 has guidance and specific requirements, equations etc.

Two important differences between the ASME B31.3 piping code and the European EN-13480 piping code are that the European code is much more strict regarding design of pipe supports that are welded to the pipe and that the European Piping Code also requires that a 3rd party analysis of piping systems and pipe supports are carried out before any assembly and commissioning can take place. This is rarely the case with fast moving offshore projects based on piping design after the ASME B31.3 piping code where 3rd party verification seldom is finished before mechanical completion and hence where any findings late in the project may lead to large modification on already installed piping and equipment. Forgotten or overseen project requirements to piping and equipment integrity during and after a blast/explosion, or fire, may lead to large modifications and redesigning of piping systems and hence large economical expenses resulting from delayed “first oil”.

Compared to the ASME B31.3 piping code, a project that entirely adapts to the EN-13480 piping code must include some headroom for additional costs related to a large amount of 3rd party verification work.

2.4.4 The Pressure Equipment Directive, PED

The European Pressure Equipment Directive, PED, is not a pressure vessel or piping design code, but it contains some essential safety requirements that must be fulfilled in the design of piping systems. If the offshore installation is going to be placed in Europe, the piping stress engineer needs to be updated on the directive and the harmonised piping codes, such as EN-13480 for metallic piping.

2.4.4.1 PED and piping materials

None of the large number of ASME piping material specifications used for decades in offshore projects in Europe are listed in the PED harmonised piping code EN 13480. The large numbers of listed European piping material specifications in EN-13480 do, however, by default meet the essential safety requirements of the Pressure Equipment Directive. If ASME materials are going to be used for an offshore installation under the PED directive, they have to be approved by an appointed PED Notified Body. As a minimum requirement the pipe and valve material specification should satisfy the PED requirements.

The trend after the introduction of PED is however that most Notified Bodies with a heavy workload on offshore projects will approve most of the ASME materials (sometimes with additional requirements) as long as they satisfy the requirement to elongation and Charpy values set forth in the pressure directive.

European countries without a large offshore industry do not seem to accept the use of ASME materials for on-shore piping systems, e.g. for piping used in European power plants, paper mills, the food industry, etc.

2.4.5 Other important design codes and standards

2.4.5.1 General

Pipe stress engineers need to be experienced in the use of many other pressure related design codes, such as pressure vessel codes and pipeline codes for subsea analysis. The reason is that the actual piping design codes do not cover all load cases, analysis techniques and calculations required to document the structural integrity of offshore piping systems.

In addition to the design codes listed above, it will be necessary to be familiar with relevant sections of the code and standards listed in the references in the beginning of this recommended practice. In the “Topside” and “Subsea” sections of this RP, a guide to relevant codes and standards are given for different calculations such as pressure thrust calculations, nozzle/shell load calculations, pump- and compressor nozzle load and alignment procedures, flange leakage calculations, fatigue calculations, blast/explosion calculations etc.

3. Topside Process Piping

3.1 General

The intention of this section is to describe what a pipe stress engineer working with offshore process piping for topside systems on Oil-and Gas platforms, FPSOs and MOUs normally has to consider prior to “stress approval” of a complete piping system. Parts of this section may also be relevant for onshore piping, e.g. oil refineries.

3.2 Commonly used design codes

The basic design code for stress engineers working with topside offshore projects is the ASME B31.3 Process Piping Code. National design codes based on the PED harmonised piping code EN-13480 have so far not been a success with regard to implementation and use in European offshore projects.

3.3 Type of calculations

3.3.1 Comprehensive calculations

Today it is no longer practical or economical to perform comprehensive analysis with other tools than specialised pipe stress software or general purpose FEA software. For further information about pipe stress analysis in general, references are made to section 2 of this RP.

3.3.2 Hand calculations

Some hand calculations will always be necessary in order to generate input data for use in pipe stress programs. If the project uses a general purpose FEA program, then a large number of hand calculations need to be performed. Such programs may lack a lot of specialised piping calculations, such as to verify combined nozzle load calculations on pumps, flange calculations, code stress calculations etc.

Any hand-calculations performed by handwriting or by use of MathCad, Excel worksheets, etc. shall have clear references to textbooks or design codes where equations are taken from, if not straight-forward and basic or well known equations are being used.

3.4 Loads to be considered in piping design

3.4.1 Dead weight

The deadweight load is the sum of weights from pipe, content, insulation, flanges, bolts, tees, bends, valve-and valve actuators etc.

3.4.2 Internal pressure

This is the static end-cap pressure load caused by the internal pressure exposed to the cross sectional area of the pipe internal diameter or for expansion joints the pipe outer diameter or mean-bellow diameter.

3.4.3 Sustained loads

Sustained loads are the sum of dead weight loads, axial loads caused by internal pressure and other applied axial loads that are not caused from temperature and accelerations etc. For ASME B31.3 the allowable sustain stress is listed in section 302.3.5.

3.4.4 Occasional loads

Occasional loads are loads such as wind, earthquake, breaking waves or green sea impact loads, dynamic loads such as pressure relief, fluid hammer or surge loads. The ASME B31.3 code has specific requirements to the accumulated hours of occurrence of such loads. For ASME B31.3, the allowable occasional stress limit is listed in section 302.3.6.

3.4.5 Environmental loads

Environmental loads are loads caused by earthquake, waves, wind, snow and accumulation of ice from sea spray or rain. Environmental loads are treated as either sustained or occasional in nature and hence should meet the stress limits for sustained-or occasional stresses.

3.4.6 Live loads

Live loads are loads that create a temporary deflection in the deck or supporting steelwork. Typical temporary deflections caused by live-loads are:

— filling or draining of a large column or pressure vessel
— material handling by landing or lifting offshore containers or other heavy equipment from a deck with piping connected to sensitive equipment supported underneath
— deck-deflections caused by bending moments from an offshore crane pedestal during heavy lift-operations (poorly designed crane-pedestal fundament)
— sag-and hog effects during loading-and offloading of oil

stored within the hull of a production vessel such as a FPSO.

Engineering judgement must be given to whether considering live loads as being added to the sustained- or displacement stress range. (Some pipe stress programmes will treat a temporary deflection as displacement stress and a temporarily load or force as sustained stress even though the source and results are the same).

3.4.7 Thermal expansion and contraction loads

Thermal expansion and contraction loads may be detrimental for the pipe itself, flanges and bolts, branch connections, pipe-supports and connected equipment such as pumps and compressors. Hot-cold system combinations of manifold piping and by-pass piping are typical examples where thermal loads have a major influence on the total stress levels. Sufficient pipe flexibility is necessary to prevent such detrimental loads.

3.4.8 Other displacement loads

Differential displacements, for example between two independently supported modules, between platforms connected with a bridge and piping running between a pipe rack and a swivel-stack (FPSO) shall also be included in calculations where relevant. Platform and module settlement may also cause stresses in the piping. Depending on the piping code being used, the stress from settlement may be defined as either sustained or displacement stress.

3.4.9 Blast/explosion loads

The ASME B31.3 code does not address accidental or exceptional loads such as blast (hydrocarbon explosion), fire, accidental heel of floaters, etc.

In contrast to ASME B31.3, the PED harmonised piping code EN 13480 does consider exceptional (accidental) design loads.

Refer section 3.11 for further information on blast load calculations.

3.4.10 Green sea

The impact loads from breaking waves or green sea entering over the shipside of a production vessel like an FPSO have to be considered in piping design. Ring main firewater piping and connected deluge skids are typically exposed to such loads. Whether these cases are actual design parameters or not are normally informed about in typical project documents such as the "Design Accidental Load Specification" or the "Environmental Load Specification".

Dynamic Load Factors should be used in combination with the design pressure for such calculations.

3.4.11 Accidental heel

Another accidental load is accidental heel of floating installations such as FPSOs, Semi Submersibles and MOUs in general. Accidental heel are caused by flooding of watertight compartments due to ship collision, corrosion, explosion and even poor design of control systems for ballasting systems. National rules and regulations, such as the Norwegian Maritime Directorate, may require that the piping should withstand a static heel of 27 degrees or a static heel of 17 degrees plus wave accelerations and dynamics. The ASME B31.3 occasional stress or the EN-13480 exceptional stress limits should be used to establish the allowable design stress for an accidental heel.

It is important to investigate whether piping without hold-downs on the guides can lift off and slide over the guide-details resulting in a large free-spanning pipe-run.

3.4.12 Accidental heat load from fire

Accidental fire is not covered by piping design codes. It is however required in some HSE related standards and codes, that pressure vessels and piping with a given volume of hydrocarbons shall be leak proof during a fire heat load for a certain period, e.g. 30 minutes, until the pressure vessels and connected piping have been depressurised to a certain level. Normally the project HSE, process-or safety department engineers will perform simplified hand calculations in order to find those piping segments (and pipe supports) that need passive fire protection insulation (PFP). The best available procedures for such calculations are given in the Scand power report "*Guidelines for the Protection of Pressurised Systems Exposed to Fire*", Report no. 27.207.291/R1-Version 2, 2004. This guideline can be downloaded from the company web pages. Refer also PED requirements to pressurised equipment listed in PED, Annex I, section 2.12,"*External fire*". The pipe stress department should be consulted by the process-or safety department if it is not possible to use PFP on piping that requires this and when the methodology outlined in the Scand power report is not suitable for the layout and mass-distribution of the actual piping.

3.5 Wall thickness calculations

3.5.1 General

Time spent on pipe wall thickness calculations according to relevant design codes are only a fraction of the total time spent on pipe stress and flexibility analysis and are normally not of any concern. Pipe stress engineers working on large projects will normally not be involved in wall thickness calculations, as the wall thickness is given in the project *Piping and Valve Specifications*. Pipe wall thickness design checks are however always by default performed by the pipe stress software and any error in the project *Piping and Valve Specifications* will then be discovered. Hand calculation of straight pipe and pipe-bends according to ASME B31.3 can be done by using the equations listed in section 304 for ordinary piping and section K304 for high pressure piping. Refer also EN 13480, part 3, section 6.1 Straight pipe and section 6.2 Pipe bends and elbows.

3.6 Flexibility calculations

3.6.1 General

A general information on flexibility analysis and code requirements are given in

— Section 2.2.3, 3.4.6, 3.4.7 and 3.4.8 in this recommend practice
— ASME B31.3, section 319 Piping Flexibility, and Appendix D, Flexibility characteristics
— EN 13480, part3, section 12, Flexibility analysis and acceptance criteria.

3.7 Equipment nozzle load calculations

3.7.1 General

This section will give guidance to relevant design codes and standards where equations and methodologies used to prove that the applied loads from the piping do not exceed allowable design loads specified by the actual codes or standard. Some piping FE programmes have such equipment design checks integrated in the software itself, others do not, and hence hand calculations according to below listed codes and standards may be necessary.

The layout of piping connected to sensitive equipment is likely to be governed by the allowed interface reaction loads. Low allowable loads may result in the increase of piping system costs while excessively high reactions may increase equipment maintenance costs. Early communication between the stress engineer, the manufacturer and the owner is essential for avoiding misunderstandings and for arriving at the best solutions for all parties.

Allowable nozzle loads for accidental loads and upset conditions need to be agreed upon. The equipment vendor should provide a set of allowable loads for these load cases in addition to and above the allowable loads given in e.g. NORSOK Standard R-001. These should also be tabulated on the General Arrangement Drawing of the equipment. One of the reasons for this is that most pressure vessel and other equipment design codes do not list allowable nozzle or shell design stresses for accidental loads.

References to standards for allowable nozzle loads and how to calculate them are listed below.

3.7.2 Compressors

Design equations for individual - and combined nozzle load calculations for compressors are given in:

API Std. 617. Axial and Centrifugal Compressors and Expander-compressors for Petroleum, Chemical and Gas Industry Services, Annex 2E.

— Additional project specification to vendor, such as to design for 2 times the API 617 Annex E values.

3.7.3 Turbines

Design equations for individual - and combined nozzle load calculations for turbines are given in:

— API Std. 611 General-Purpose Steam Turbines for Petroleum, Chemical and Gas Industry Services
— API Std. 616 Gas Turbines for Petroleum, Chemical, and Gas Industry Services
— NEMA SM23 Steam Turbines for Mechanical Drive Service
— Additional project specification to vendor, such as 2-4 times the NEMA SM23 allowable.

3.7.4 Turbo-compressors

Design equations for individual - and combined nozzle load calculations for turbo-compressors are given in:

— API Std. 617. Axial and Centrifugal Compressors and Expander-compressors for Petroleum, Chemical and Gas Industry Services, Annex 2E
— Additional project specification, such as 2 times the API 617 Annex 2E values.

3.7.5 Centrifugal pumps

Design equations for individual - and combined nozzle load calculations for Centrifugal API Pumps are given in:

— API Std. 610 Centrifugal Pumps for Petrochemical and Natural Gas Industries
— Any additional project specification, such as 2-4 times the API 610 allowable (Refer NORSOK Standard R-001, Mechanical Equipment, section 6.2.2).

3.7.6 Pressure vessels

Pressure vessels for topside process systems are such equipment as Separators, Knock-Out Drums, Heat- Exchangers, Scrubbers etc.

The nozzle loads from connected piping shall not cause stresses in the nozzle-to shell interface to be larger than allowed by the actual pressure vessel code. Typical pressure Vessel codes are:

— ASME BPVC Section VIII, Rules for Construction of Pressure Vessels
— PD 5500 Specification for Unfired Fusion Welded Pressure Vessels
— EN 13445 Unfired Pressure Vessels.

Normally the allowable nozzle loads will be tabulated on the vendors *General Arrangement Drawing* of the actual pressure vessel. Some standards, like the *NORSOK Standard R-001 Mechanical Equipment*, have equations for allowable nozzle loads that the operator and vendor can agree upon in a specific project.

For the NORSOK Standard R-001, these nozzle load equations are given in section 5.1.5 of that standard.

For large diameter vessels credit can be taken for nozzle stiffness using the “Nozzle Flexibility Method” as described in M. W. Kellogg, “Design of Piping systems” or other recognised literature.

3.7.7 Skids with piping connections

General arrangement drawings of prefabricated skids with internal piping, such as *Oil and Gas Metering Skids,* should contain tables with allowable flange-or nozzle connection loads. Such loads can be based on flange leakage criteria or e.g. *NORSOK Standard R-001 Mechanical Equipment*, *section 5.1.5*. In addition to allowable nozzle loads, the thermal expansion of the interface flange-or nozzle connection should be tabulated on the general arrangement drawing.

All skid pipe-nozzles should be fully restrained within a distance of 3 pipe diameters from the flanged end and all thermal expansion of the skid internal piping should be taken within the skid itself.

Where it is unpractical for the skid vendor to contain all thermal expansions within the skid itself the skid piping should be modelled together with the external pipe in a pipe stress programme.

3.7.8 Manifolds

In those cases where the engineering and fabrication of large manifolds are performed by a subcontractor etc. it will be necessary to agree upon the allowable nozzle-or flange connection loads. Connections may also be welded. The same load criteria that apply to skids can be used as a basis. All manifolds should be modelled in a pipe stress programme according to fabrication drawings with all external piping connected. Hot-and cold system combinations from the influence of connected piping shall be evaluated. In those cases where allowable loads are exceeded, a flange loading check according to the relevant flange connections in combination with an allowable code stress check of the manifold header should be sufficient to document the integrity of the manifold and connected piping. Manifolds may be constructed with basis in piping design codes or pressure vessel codes.

3.7.9 X-Mass Trees

Allowable nozzle loads for flowline connections should be equal to the force and bending moment capacity of the interface API connection and shall in general be agreed with the subcontractor.

3.8 Flange calculations

3.8.1 General

ASME B31.3 Process Piping Code specifies that piping systems shall have sufficient flexibility such that leakage at joints (flanges) is prevented due to external applied loads and internal pressure and temperature.

Companies that do not have a pipe stress programme that include a flange-check calculation module, should document the flange capacity by hand-calculations or use of dedicated flange-check programmes.

ASME B31.3 does not specify a calculation procedure or methodology, so there is no strict requirements to the flange calculations, but it should be a well recognised method commonly used by pipe stress societies. It should also be mentioned that the methodology used in most of the codes and standards referred to below can not guarantee a 100% leak-tight flanged connection, but the use of these codes and standards should prove that the stresses and deformations of the flange components including the bolts should be within code allowable values.

EN-13480 Metallic Industrial Piping Code. This code points to a pressure vessel code (EN 13445) which again points to the most reliable code for flange leak-tight calculations available, the European *EN-1591.1 Flanges and their joints. Design rules for gasketed circular flange connections, Calculation method.*

There are also German Nuclear Standards KTA3201.2 and KTA 3211.2 that in addition to the EN-1591.1 code above includes the effect from torsion of the flanges. The use of these codes (EN1591.1 and KTA) do however demand a lot of data and information about the gaskets being used, and these data should preferably be obtained by physical tests.

3.8.2 ASME B16.5 flange calculations

The following methodologies are commonly used:

a) Start with the well known “Pressure Equivalent Method” as described in *M.W. Kellogg, ”Design of Piping systems*” or in ASME Section III, Division 1, NC-3658.1. (The latter is also the only code that addresses a methodology for flanges under accidental/exceptional loads such as a blast condition). Make sure that external axial loads trying to separate the flanges are included. There are also variants of the “Pressure Equivalent Method” in use by some Engineering Companies. One of these variants is the “*Blick Theory*” from the 1950’s. Refer also “*CASTI Guidebook to ASME B31.3 Process Piping*”, *chapter 2*. A piping system designed according to ASME B31.3 shall have the flanged connection hydrostatically tested to 1.5 x design pressure and it has therefore been a practise among many large Engineering Companies to allow the total equivalent pressure including external bending moments and axial forces to reach a level of 1.5 times the rated flange pressure at temperature.
b) If calculations according to the traditional “Pressure Equivalent Method” fail, then try the methodology outlined in ASME PBVC Section VIII, Division 1, Appendix 2.
c) If calculations to ASME VIII, Div1, Appendix 2 also fail, the last option will normally be to use FE-analysis. This does, however, demand a lot of FEA skill from the person performing the calculations.
d) If none of the above calculations can validate the flange installed, try to modify the piping layout, adjust some supports or try to change the flange pair with a higher rated flange pair, e.g. #600 instead of #300. The last option may be to weld the pipe instead of using flanges.

In those cases where some parts of the piping are designed according to ASME B31.8 instead of ASME B31.3 (e.g. piping between the riser and the pig receiver) the flange leakage criteria given in ASME B31.8, Note (14), can be used.

3.8.3 API Flange calculations

The API flange standards consider high-pressure piping flanges with ring-type-joint metal sealing rings.

Documentation of API flanges capacity to withstand external loads in combination with internal pressure should be performed by using the methods and charts outlined in *API 6AF, Capabilities of API Flanges under Combination of Load*.

3.8.4 Clamp Connections

High pressure piping spools used in drilling units, flowlines and for water injection piping are often connected to each other by hub-clamp type connections. Manufacturers of clamp connections have to design the clamp connections according to ASME VIII, Div 1, Appendix 24 “Design Rules for Clamp Connections”. The commonly most known connection is the “Gray lock-type” coupling. Connections shall be documented to have bending moment within vendor allowable (found in the vendor product catalogue). However, most of these clamp couplings have the strength comparable to a welded joint, and for screening purposes it is common only to check that the pipe stresses at the hub-welds are within e.g. 80% of the piping code allowable stress.

3.8.5 NORSOK Compact Flanges

NORSOK high strength Compact Flanges types NCF5 are confirmed to apply with the European Pressure Directive by Det Norske Veritas.

These flanges allows for much higher external loads than the ASME B16.5 flanges, still being smaller and with reduced weight compared to traditional flanges. The required bolt prestress do however not comply with the ASME B31.3 piping code allowable bolt stress.

Details about these flanges are given in *NORSOK Standard L-005 Compact Flanged Connections*. (An ISO Standard, ISO 27509, is pr 2008 under construction and will have references to NORSOK L-005 when finished).

3.9 Pressure relief- and discharge force calculations

3.9.1 General

Pressure relief- and discharge reaction forces applied from the relieving device to the piping system are dynamic in nature and have a short time of duration. Hence the piping may see larger loads than those produced under static application of the load. Equations used should therefore include a Dynamic Load Factor, DLF. If the DLF is not based on modal analysis, then a conservative value in the range 1.5-2.0 shall be used.

3.9.2 Pressure Safety Valve discharge reaction forces

Design equations for pressure relief or discharge of steam, gases and fluids are given in the following codes and standards:

— ASME B31.1 Power Piping, Appendix II
— API RP 520 Sizing, Selection and Installation of Pressure-relieving Devices in Refineries
— EN 13480, part 3, appendix A.2.4. (This appendix also includes a DLF selection figure).

Refer also *“CASTI Guidebook to ASME B31.3 Process Piping*”, chapter 3, for selection of DLF.

3.9.3 Bursting-and rupture Disc reaction forces

Design equations for pressure relief or discharge of steam, gases and fluids are given in the following codes and standards:

— API RP 520 Sizing, Selection and Installation of Pressure-relieving Devices in Refineries.

3.9.4 Flare-tip reaction forces

These loads are given on vendor data sheets. They should be multiplied by a DLF if not confirmed to be included in the reaction force listed on the data sheet.

3.10 Expansion and slip-joint thrust load calculations

3.10.1 Expansion joint thrust load

The description below is for axial expansion joint of bellow type without tie-rods. There is a large variety of expansion joints of the bellow type, i.e. axial bellows with tie-rods, angular bellows with hinges etc.

- Refer to the EMJA Standards for design, installation and use of expansion bellows.
- Refer also to EN-13480, part 3, section 6.5, for piping to be installed in Europe.
- If no vendor information is available regarding the effective pressure trust area of the bellow, then the mean bellow diameter should be used in the calculation for the cross sectional area where the pressure is applied. This area is always greater than the area represented by the pipe's outer diameter.
- There must be a strong anchor-or line stop on either side of the broken pipe to take the large pressure thrust forces. Bellow spring force and friction forces from sliding pipe supports should also be included in the pipe support anchor calculations.
- One of these line-stops or anchors should be as close to the expansion-joint as practical. Piping should be guided with spacing according to vendor specifications.
- The use of expansion-joints for offshore installations should be minimised and always be approved by the owner/operator.
- Expansion-Joints should never be applied to ring-main firewater or piping with hydrocarbons that are going to be designed for an accidental blast.

3.10.2 Slip- joint thrust load

Slip-joints are commonly used for cargo-piping systems on oil tankers. They are small in size and can take large pipe axial movements arising from temperature variations, loading-and offloading and vessel deck-deflections do to sag-and hog of the hull due to large waves passing underneath the vessel.

- The pipe's outer diameter should always be used in the calculation for the cross sectional area where the pressure is applied.
- There must be a strong anchor-or line stop on either side of the broken pipe to take the large pressure thrust forces. Bellow spring force and friction forces from sliding pipe supports should also be included in the pipe support anchor calculations.
- One of these line-stops or anchors should be as close to the slip-joint as practical. Piping should be guided with spacing according to vendor specification.
- The use of slip-joints for offshore installations should be minimised and always be approved by the owner/operator.
- Slip-Joints should never be applied to ring-main firewater or piping with hydrocarbons that are going to be designed for an accidental blast.

Extreme care should be carried out when a mixing of so-called internally axial restrained- and open (no internal axial restraining) couplings of type "*Straub pipe couplings*", which are not pressure-balanced, and similar designed couplings are being used on the same line. Although not advised, the first may be used without pipe line stops or anchors, whereas the latter must have pipe line-stops or anchors in order to take the end-cap forces from the internal pressure.

Extreme care shall also be provided if expansion-and slip-joints are being used on piping systems that should maintain their integrity after an accidental blast. Typical systems are firewater ring main and piping containing large amount of hydrocarbons. The reason is that it is difficult to calculate the differential displacements between the pipe support anchors or line-stops after an explosion and that such calculations have to be accurate within e.g. 50-100 mm differential displacement of the pipe support anchors in order to maintain the integrity of the joint. Hence expansion-and slip-joints should not be allowed on piping that have to maintain the integrity after an explosion.

3.11 Blast load calculations

3.11.1 General

Evaluation of the structural integrity of piping and pipe supports during, and after, an accidental blast, or any other exceptional event, should always be performed by comprehensive methods such as FE pipe stress analysis. It should be noted that international research work is ongoing in order to come up with a method for the blast effect on piping that takes both the blast overpressure and the dynamic drag load into consideration. For smaller pipe sizes there will only be the load from the dynamic drag pressure, bur for larger pipe diameters and pressure vessels there will also be a component from the overpressure in front of the blast. There is no international agreement on the exact pipe sizes where elements from the overpressure should be considered. The methodology to calculate the total reaction forces on the piping is also outstanding. This RP will be updated with a more comprehensive design philosophy when the international research work has concluded. Hence this RP only considers the traditional method which is to ignore the overpressure and design for the dynamic drag pressure alone. For further information it is recommended to visit the FABIG web-pages.

For accidental and/or exceptional load cases not covered by the codes the stress engineer/safety engineer should agree on a set of rules and limitations together with the owner and the third party verificator. Modification projects may include a variety of existing piping systems and equipment which differ largely with regards to what loads they have been designed for previously. Some modification projects consist of tying in new pipe to systems never designed for a blast scenario. For these projects an agreed blast procedure will benefit all parties.

3.11.2 Accidental blast loads and allowable stresses

Pipe stress blast calculations should be performed in two steps:

a) Blast wind calculations

   Calculation of pipe stresses, support and equipment nozzle loads due to the blast wind (dynamic drag pressure) alone.

   - ASME B31.3: The stress limit for this event will be the occasional stress limit = 1.33 x Basic allowable stress at temperature. (The PED harmonised piping code EN 13480 allow for 1.8 x basic allowable stress at temperature. The same utilisation should be allowed for in PED projects where the ASME B31.3 piping code is used instead of the harmonised piping code)
   - EN 13480: The stress limit is 1.8 x basic allowable stress at temperature.

b) Impact from deformation of structural steelwork

   Pipe stress calculations, flange integrity and equipment nozzle loads due to permanent displacement of pipe supports connected to primary and secondary steelwork that undergo a plastic deformation have to be performed. Typical are decks at different levels that deform permanently in opposite direction of each other due to the blast overpressure.

   - ASME B31.3: The allowable stress for this event is higher than for the blast wind as this case can be considered analogue to a pipe settlement scenario, ref ASME B31.3 para. 319.2.1 (c). Hence an allowable stress equal to the "liberal equation", ASME B31.3 para. 302.3.5 (d) (1b) should be allowed for. Alternatively the allowable operating stress as outlined in ASME B31.3, Appendix P, can be used. This alternative rule allows for a stress range up to 2.5 x basic allowable stress at moderate temperatures.
   - EN 13480: For PED projects no stresses can exceed the allowable stresses calculated according to EN 13480, section 12.3.6. (Minimum of 3 x basic allowable stress or 2 x proof strength at temperature).

3.11.2.1 Blast drag pressure

A conservative rule of thumb in early design studies is to assume that the drag pressure is 1/3 of the blast overpressure. This gives conservative estimates up to an overpressure of approximately 2 bar. Thereafter the thumb of rule is non-conservative.

Table 3.1 below shows actual wind speeds that result in the same pipe drag loads as the dynamic drag pressure listed in the same table. The equivalent wind air-density used in calculations below is 1.224 kg/m^3 which are a default density of air used by most pipe stress programmes in calculations of wind loads. (A sudden wind-gust of 20 m/s will knock a person over. This equals a dynamic drag pressure of only 0.0025 bar).

If no other information is available, blast drag loads should be considered to occur from all main directions, also downwards, e.g. through deck-grating. It is however not required to analyse two independent blast events happening at the same time, e.g. due to two ignition sources perpendicular to each other.

In order to obtain more reliable values for the overpressure and drag pressure for an offshore installation with steelwork, piping, firewalls, equipment etc a 3D blast event simulator should be used. The FLACS (FLame ACceleration Simulator) programme is a commonly used software tool for such simulations.

Table 3-1 Blast drag pressure equivalent wind speeds.

Explosion Overpressure [bar]	*Dynamic Drag Pressure [bar]*	*Equivalent Wind Speed for Drag Load Calculations [m/s]*
0.2	0.02	57
0.4	0.05	90
0.6	0.10	128
0.8	0.15	157
1.0	0.21	185
1.2	0.28	214
1.4	0.36	243
1.6	0.45	271
1.8	0.55	300
2.0	0.65	326
2.2	0.76	352
2.4	0.88	379
2.6	1.00	404
2.8	1.13	430
3.0	1.27	456
3.2	1.41	480
3.4	1.57	506
3.6	1.73	532
3.8	1.88	554
4.0	2.05	579

Tabulated pressures are based on figure 5.3 in *FABIG Technical Note No 8, Protection of Piping Systems Subjected to Fires and Explosions.* This table is only meant for information and to show that the relationship between overpressure and drag pressure is not linear.

Typical estimates of blast overpressure for a variety of offshore installations and areas are listed in DNV-OS-A101 *Safety Principles and Arrangements*, *table D1*.

3.11.2.2 Blast drag load

Hand calculations of the blast drag load per unit length of the pipe can be calculated as follows:

$$F_D = 1/2 \cdot \rho \cdot v^2 \cdot D \cdot C_D \cdot DLF$$

where

F_D = Drag load from the blast pr unit length (N/m)
ρ = density of the combustion gases (kg/m^3)
v = velocity of the combustion gases (m/s)
D = pipe diameter including insulation (m)
C_D = Drag coefficient for blast = 1.0 (Ref. 3.11.2.5)
DLF = Dynamic Load Factor (Ref. 3.11.2.6)

The term $1/2\rho v^2$ equals the dynamic drag pressure used to calculate the equivalent air velocities in table 3.1 above.

3.11.2.3 Blast and operational flexibility

It should be noted that it is difficult and time-consuming to design for a blast drag load higher than 0.5 bar, especially for piping that requires flexibility in the operating condition, such as pipe spools close to gas-compressors and turbines that require low nozzle loads and hence very flexible piping that includes use of spring supports. Piping around such equipment may have to be designed with a double set of pipe supports, one for the operating condition and one for the accidental condition. Double sets of pipe supports should however be agreed with the operator in order to ensure that supports for the blast conditions are not mistakenly treated as operational supports. This could be a serious safety issue during normal operating condition, e.g. if required gaps on a guided blast support are overseen and closed by the welder. Hence blast supports should be painted with a specific colour.

3.11.2.4 Blast and structural deformation

Structural steel-work is by most design codes allowed to have plastic hinges and hence structural steelwork that pipe supports are welded to, may see large plastic deformations. Considerable structural deformation and equipment movement during explosion shall be taken into consideration in the piping design. This may require that some pipe supports may have to be designed with a weak-link, or breaking pin, that breaks when the structural steelwork deflects more than a certain limit and hence protects the piping from being overstressed and piping to leak. Typical situations are vertical piping being supported at two different deck levels where the two decks deflect in opposite directions during the explosion.

3.11.2.5 Blast drag coefficient, Cd

There is an international understanding that the drag coefficient for the blast combustion gases can be much higher than normally selected by use of typical steady-and turbulent flow diagrams and Reynolds numbers. The tabulated blast equivalent wind-velocities would by use of such diagrams typically result in Cd values for piping in the range 0.6-0.7. These diagrams are, however, not valid for a fluid that accelerates, such as a blast wind.

API RP 2F, Recommended Practice for the Design of Offshore Facilities against Fire and Blast Loading, recommends that a drag coefficient, Cd = 1.0, should be used for blast load calculations of piping, and hence a Cd = 1.0 is the recommended value for blast design in this RP.

3.11.2.6 Dynamic Load Factor for blast, DLF

If no dynamic analysis or modal analysis to find the DLF for use in static analysis is carried out, then a conservative DLF in the range 1.5-2.0 should be used in order to account for the dynamic effect of a blast.

Dynamic Load Factors which are closely linked to the natural frequencies of the pipe span can be found in the British Steel Construction Institutes *Document No. 209 "Interim Guidance Note, Section 3, Design Guidance for Explosion Resistance, 1992*".

For a triangular pulse which starts at zero and reaches a maximum value at 30% of the total blast duration time (t_d), the maximum response as a function of rise time to natural period (T) is shown in Figure 3.1 below.

Example:

In a given project, the DAL report (Design Accidental Load Specification) report, states that figure 3.13 in the *Interim Guidance Note* referred to above should be used. This figure is shown below as Figure 3.1. Further, the DAL specification tells that the blast duration time, t_d, = 0.15s should be used in the project.

From modal analysis the pipe stress engineer has found that the first mode of vibration for a given pipe span has a natural frequency, f_n, = 20 Hz corresponding to a periodic time, $T = 1/f_n = 1/20 = 0.05s$. The Dynamic Load Factor are then found by calculating the t_d /T ratio = 0.15/0.05 = 3.0. From Figure 3.1 below it can be seen that the corresponding DLF is approximately 1.0 which is far less than a conservative value of 1.5-2.0 chosen as default for all piping systems when modal analysis is not used.

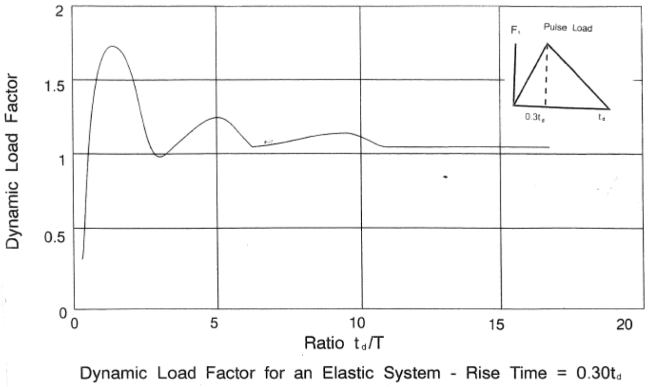


Figure 3-1
DLF selection figure used in example above

3.11.2.7 Temperature and internal pressure effects

The allowable pipe design stress and Young's Modulus for the blast condition should be adjusted to reflect the design temperature and not the properties at room-temperature. Blast analysis should be carried out in combination with the total deadweight of the piping and the internal design pressure. Thermal expansion stresses will normally not contribute much to the total stresses as this is a secondary self limiting stress. Thermal expansion stresses are therefore often ignored in blast analysis.

3.12 Fatigue calculations

3.12.1 General

The pipe stress and flexibility analysis should normally be extended to a formal or simplified fatigue analysis when there is more than one additional cyclic load source of importance to the expansion-and contraction or alternating bending stresses of a piping system, e.g. other sources than pure temperature cycles which is taken care of by the equation for displacement stress calculations in most piping design codes and thereby automatically accounted for in the "code check" section of commonly available pipe stress programmes. Design and construction must ensure that due consideration is given to the risk of fatigue due to vibrations in pipes. A Modal analysis of all piping systems should be performed and it is desirable and a common practice to keep the piping system's natural frequency above 4Hz to mitigate circumstances where fatigue can be induced by low frequencies of vibration. For situations where large expansion loops are required to absorb large movements, natural frequencies above 4Hz may be difficult to achieve

3.12.2 Vibration

Vibration effect on the fatigue life of the piping is to be examined when piping is connected to machinery such as reciprocating pumps and compressors. Blow-down and flare piping where high gas velocities are expected may be exposed to high frequency (acoustic) fatigue. One usual way to overcome acoustic fatigue is to increase the pipe wall thickness for a calculated specific length.

Vibration caused by wind induced vortex shedding is described in Appendix A of this recommended practice.

Recommended procedures used in order to screen and avoid vibration-caused fatigue in piping are given in *Guidelines for the Avoidance of Vibration Induced Fatigue in Process Pipework, Second edition March 2008. Published by the Energy Institute, London.(ISBN 9780 852934630).*

3.12.3 Typical piping exposed to fatigue

Piping systems that by default should be analysed or evaluated for fatigue damage are:

— Piping connected to surface wellhead or Xmas trees where flexible hoses or "chiksan" type couplings are not used to take the vertical and horizontal movements.
— Piping along a bridge between two platforms, especially at the sliding landing area of the bridge.
— Piping running along the deck of a FPSO or in pipe racks along the FPSO that are subjected to vertical sag- and hog deflections from loading, offloading and waves. If no specific project data is available, a longitudinal compression-and expansion of +/- 10 mm per 10 m pipe from the piping fixed point should be used for initial design. Actual design values must be verified later in the project.
— FPSO piping-spools with large unsupported overhang, poorly supported valves and valve-actuators, etc. that are subjected to vessel accelerations from sea actions (heave, pitch and roll accelerations).
— Piping connected to reciprocating pumps and compressors that induce low forced frequencies that could coincide with the natural frequency of the piping system.
— Thin walled duplex steel piping exposed to high gas velocities, so called acoustic fatigue.

3.12.4 Fatigue analysis of wellhead flowlines

Wellhead flowlines must be subjected to a comprehensive fatigue analysis given the fact that they are exposed to high cycle loadings from Xmas tree movements and flow induced vibrations. As ASME B 31.3 only to a limited degree takes into account fatigue damage, a more detailed fatigue calculation according to PD5500 should be performed for all flowlines and gas lift lines.

Wave loadings at the conductors initiate cyclic loads at the Xmas trees and an additional fatigue check should be performed.

The flow induced loads with the largest contribution to the fatigue life are not the design slug loads, but rather loads generated by minor density fluctuations in the well stream. These flow induced loads are not applicable for gas lift or water injection lines.

According to PD5500, the fatigue assessment shall be based on the primary plus secondary stress category and the full stress range is to be used.

The design lifetime of flowlines and gas lift lines should be 30 years unless otherwise specified by Company. Frequent inspection should be initiated after 1/3 of the estimated lifetime.

Frictional effects in pipe supports may be significant in fatigue analysis since they tend to increase the system resistance to Xmas tree movements, ref EN13480 12.2.10.3.1. These effects tend to be of importance in systems where there are one or more supports (not spring only) relatively close to the Xmas tree and where no line stops prevents the lines from moving relatively to the other supports.

When preliminary evaluations indicate that frictional effects may be of significance, these effects shall be more thoroughly investigated through incorporation in the fatigue load cases.

In order to do a representative analysis, the analyst should bear in mind that the fatigue wave loads (Xmas tree movements) represents oscillations around a steady state deformation pattern created by well growth, weight, temperature and pressure. Hence, the friction loads should be based upon the reaction loads from these steady state solutions.

The results from fatigue analysis should be reported in the pipe stress report. Any spool not meeting the 30 years fatigue life design requirement shall be identified. Design life time to be stated on the Stress Isomeric Drawing for the pipe.

3.12.5 Recommended design codes and standards for fatigue analysis

a) For investigation of the fatigue effect from wave loading alone, the methodology listed in *DNV RP-C203 Fatigue strength analysis of offshore steel structure* can be used. It is however not common to include *Design Fatigue Factors*, *DFF*, in topside piping design, but it might be used for piping that is insulated and otherwise difficult to, or seldom, investigated for cracks and corrosion. DNV RP-C203 is mainly intended for steel structures and not piping that in addition to wave induced deflections and accelerations will see a number of other fatigue sources such as temperature variations, pressure transients, slugging, live load cycling etc.

b) The Institute of Gas Engineers Code*, IGE/TD/12, Pipework Stress Analysis for Gas Industry Plant.* This code can be used for evaluation of high frequency fatigue caused by high gas and steam velocities, so called acoustic fatigue.

c) Acoustic fatigue: *CONCAWE Report 85/52, Acoustic Fatigue in Pipes* and *NORSOK L-002, Appendix A.*

d) The general recommended design procedure for fatigue analysis of piping systems is described in the British Pressure Vessel Code *PD5500 Specification for Unfired Fusion Welded Pressure Vessels*, *Annex C*. PD5500, working example W.6.2.3, table W.6-4, W.6-5 and W.6-6, contains an easily understandable and conservative methodology for including all fatigue loadings (imposed movements, pressure transients, thermal gradients, etc.) based on the well known Miner-Palmgren fatigue damage calculations.

If the methodology outlined in PD5000 is going to be used for subsea piping and topside part of riser-or export flow lines, then the design fatigue factor, DFF, as described in DNV-RP-C203 should be used in the fatigue life calculations. The methodology outlined in PD5500 can be used in combination with fatigue S-N curves (and their tabulated values) for welded details taken from various design codes such as ASME VIII, PD5500 and DNV-RP-C203.

An example of a fatigue calculation for a line running along a bridge that spans between a riser -platform and a production-platform in the North Sea is shown in Appendix J in this RP. The bridge is fixed to one of the two platforms and is sliding in the landing area on the other platform. This in order to allow for relative movements between the two platforms from wave loading. In addition to wave loading the most stress utilised pipe bend in the bridge landing area will see loads from slugs, temperature gradients and pressure fluctuations. The fatigue calculation presented in Appendix J is based on PD5500 Annex C, and working example W.6.2.3. in Appendix W in PD5500.

3.13 Non-standard component calculations

Piping components that are not made in compliance with any of the 100 piping component standards listed in ASME B31.3, Table 326.1 “Component Standards”, shall be demonstrated to have sufficient mechanical integrity and documented as a ‘SPECIAL ITEM’ with a Data Sheet according to the procedures outlined in ASME BPVC Section VIII, Division 2, part 4 or 5. Typical components are special flange-or hub connections and special branch-connections. Refer section 5 in this RP for further details (procedures, documentation, checklist etc).

3.14 Load case description- and combinations

A logical description of each load case shall be listed in the pipe stress report. Load cases and their combinations, both code required, and those evaluated important by the stress engineer, shall be listed in a table or matrix that are easily understood. Appendix G and H show some project examples of typical load case combinations and matrixes.

3.15 Pipe stress priority piping

3.15.1 General

In section 3.15.6 below is a list commonly referred to as “pipe stress critical line selection list”. This list was originally made by the NORSOK committee during the work with the NORSOK standard L-002 *Piping design, layout and stress analysis*. It was based on similar selection lists at that time used by 5-6 oil and gas companies. Selection criteria was then formalised in NORSOK L-002 in order to have one common selection list for all piping installations that were going to be placed in the North Sea.

3.15.2 Conflict with piping code requirements

It has been a practice for decades that only piping that falls into the critical line selection criteria below or similar lists are analysed by use of specially pipe stress FE computer software based on the beam element method. Such selection lists result in that only 35-50% of topside process piping is subjected to comprehensive pipe stress and flexibility analysis carried out by qualified personnel, and that 50-65% of all installed topside pipe-work in the North Sea lacks documentation regarding piping code requirements to flexibility analysis. (Refer ASME B31.3 para. 319.4.2 and EN-13480 section 12.2.10.1 for code requirements to flexibility analysis).

3.15.3 Visual approval of piping systems

Visual approval of piping systems is only acceptable for piping that is more or less a 1:1 duplicate of piping systems analysed by use of comprehensive methods such as FE-analysis, or for piping systems that can be documented to have had a successful history of operation for at least 10 years without any documented analysis. It should be noted that a history of successful operation not necessarily qualifies the piping to withstand long term low cycle fatigue or accidental events that have not yet taken place, such as an earthquake and hydrocarbon explosion that the piping and its supports should have been designed for. In ASME B31.3 visual approval of piping systems is described as “approved by comparison”.

3.15.4 Requirement to documentation of visually approved piping

Visually approved piping systems should be listed in a separate section in the pipe stress report with the following information:

a) A statement telling that the pipe size, material, layout, supporting, pressure, temperatures, imposed deflections, environmental and accidental loads are more or less identical to the visually approved system and the reference piping system that has been analysed by comprehensive methods.

For the reference piping system (the one being duplicated) the following information should be given:

b) Name of project and installation for which the reference piping system has been installed
c) The pipe stress report document number
d) The exact pipe stress computer file name
e) The pipe stress isometric ID
f) The relevant line numbers
g) From where (organisation or project) a copy of the documentation for the reference piping system can be obtained.

3.15.5 Critical line selection list

Due to the fact that only 35-50% of all piping systems in the North Sea can be documented to meet the piping codes requirement to pipe stress and flexibility analysis, DNV has chosen to slightly modify and rename a typical traditional "critical line selection list" taken from the NORSOK Standard L-002, Rev. 2, 1997, that has been used in the North Sea for a decade, to "Pipe Stress Priority Piping". A few items have been added to the original NORSOK Standard L-002, Rev. 2, 1997 critical line selection criteria. This implies that piping belonging to one or several of the below listed criteria should always be given priority in the pre-, basic-and detail engineering phases of the project. Piping that is sorted out based on the selection criteria for "pipe stress priority piping" should be listed in a pipe stress critical line report.

Those lines that do not fall into these criteria should however, in some way, be documented to meet the code requirement to sufficient pipe stress flexibility. Such documentation should always be finished and reviewed by a stress engineer before oil-and gas production or start-up takes place. The "Pipe Stress Priority Piping" selection criteria are listed below.

3.15.6 Pipe Stress Priority Piping

The following selection criteria apply:

a) All lines at design temperature above 180°C.
b) 4" NPS and larger at design temperature above 130°C.
c) 16" NPS and larger at design temperature above 105°C.
d) All lines with design temp. below -30°C and where the largest possible ΔT >190°C.
e) Lines 4"and larger with design temp. below -30°C and where the largest possible ΔT >140°C.
f) Lines 16" and larger with design temp. below -30°C and where the largest possible ΔT >115°C.

Note:

The ΔT temperatures in d), e) and f) above are based on a design temperature of 30°C above maximum operating temperature. When the maximum design temperature is defined to equal the maximum operating temperature then the ΔT values above should be reduced by 30°C.

---e-n-d---of---N-o-t-e---

g) Lines 3" NPS and larger with a wall thickness larger than 10% of the outside pipe diameter. (Typical are water injection piping and high pressure API piping used in drilling units).
h) Thin walled piping of 20" NPS and larger with wall thickness less than 1% of the outside pipe diameter (typical is gas turbine power generator exhaust piping).
i) All lines 3" NPS and larger connected to sensitive equipment such as rotating equipment. However, lubrication oil lines, cooling medium lines etc. for such equipment shall not be selected due to this item.
j) All piping expected to be subjected to vibration due to internal and external loads such as pressure transients, slugging, vortex shedding induced oscillations, high gas velocities and herby acoustic vibrations of the pipe wall membrane.
k) All piping connected to pressure relief valves and rupture discs.
l) All blow-down piping 2" NPS and larger excluding drains.
m) All piping along the flare tower.
n) All piping above 3" NPS likely to be affected by movement of connecting equipment or by structural deflection.
o) GRE piping 3" NPS and larger.
p) All piping 3" NPS and larger subject to steam out.
q) Long vertical lines (typical 20 meters and higher).
r) All production and injection manifolds with connecting piping.
s) The ring-main firewater line including the deluge headers and all hydrocarbon lines containing oil and gas if the installation is going to be designed for a safe shut-down after an accidental design blast/explosion. Any other lines defined in the project DAL specification or similar project document to be intact after an explosion. (Check national regulations and any relevant class rules such as DNV-OS-A101).
t) Lines falling into Category III according to PED for installations going to be placed in Europe.
u) Other lines requested by the owner, class society, the project or responsible pipe stress analyst to be critical.

3.16 Design of pipe supports

3.16.1 General

Design of pipe supports is normally carried out by the "Pipe Support Department" in large Engineering companies, or by the "Structural Department" in smaller Engineering Companies. It is not common that the pipe stress engineer does both the pipe stress analysis and the pipe support design. This would normally be a waste with of available resources.

The loads and deflections that the pipe-supports are designed to withstand are given to the pipe support-or structural department from the pipe stress analyst. Pipe support details that are going to be welded to the pipe itself, such as a pipe support shoe, shall normally be designed to meet the stress limits of the piping code, whereas the part of the support that is frame-work and guide-or hold-down details, normally are designed to meet the criteria of a Structural Steel Design code.

3.16.2 Pipe support rigidity

It is of major importance that the pipe stress engineer communicates with the pipe support- or structural department if there are any pipe supports that have to be absolute rigid. The reason is that pipe supports subjected to large loads may be within code allowable stress limits, but the deflection may be too large and not acceptable for the pipe stress analyst. E.g. calculated loads on sensitive equipment nozzles may be unrealistic and too low if the stress engineer assumes that the pipe supports are totally rigid. If the pipe stress engineer has special concerns about the rigidity of a specific support, the permitted deflection under load should be stated on the stress isometric. Supports positioned to protect sensitive equipment should be designed as adjustable supports.

Pipe-support restraints, such as guides and line-stops, are in the pipe stress programmes by default set to be totally rigid. Most available pipe stress programmes have the possibility to model the whole pipe support- steelwork or include the translational and rotational stiffness of the restraints when known. This should always be performed for supports close to sensitive equipment, e.g. the last line stop and the 2-3 last guides before a compressor nozzle.

3.16.3 Local stresses in pipe from a trunnion, lugs and other local attachments

Local stresses in the pipe wall should be checked for all stress critical lines, regardless of trunnion length or diameter. Traditional hand calculations according to the well known 'Kellogg Line Load' method or the methodology outlined in EN-13480, part 3, should be used for trunnion calculations. (The 2002 edition of the EN 13480 code, section 11, contains some errors, e.g. are some plus signs missing, in some of the equations for summary of stresses).

In addition to EN-13480, the WRC bulletin no 198 and the later 448 covers local stresses from lugs and other rectangular attachments. It is however recommended to perform FE analysis of such attachments if they are considered critical to the application, e.g. thin walled piping with high radial load from attachment.

3.16.4 Pipe support friction

When a pipe expands due to increase in temperature from the installed condition, some friction forces will be transferred to the line-stops and rest supports tending to move them in the same or opposite direction of the thermal expansion. It is advisable that the pipe stress analyst do not list the friction forces in the load table on the stress isometric. The reason is that this has led to that the friction load has been misunderstood by the pipe supporter or structural department, and hence a line-stop has been installed where the pipes were to move freely in the axial direction. Guidance should be given to the pipe support department on how to handle friction forces not listed on the stress isometric. Typical is to add an axial friction load of 1/3 of the reported vertical loads. The pipe stress analyst should always consider friction forces from pipe supports when nozzle load reaction forces are being analysed.

The appliance of friction should always be considered in conjunction with the actual pipe movements. If the movements are considered to be low (less than 3-5 mm) the effect of friction will normally be negligible. If the friction is applied without careful consideration, it can lead to unnecessary support structures.

When low friction is required, special low friction pads can be bolted or glued to the frame on which the pipe or pipe support-shoe rests on. Glued low friction pads do however have a tendency to loosen over time and should therefore be inspected periodically.

3.16.5 Bracing of branch connections

Piping branch connections in services that give potential for piping vibration, should be designed with bracing. Unsupported branch connections with a mass concentration (e.g. high vent and low drain valves) attached should be braced against the parent pipe for the following services:

— process rotary compressor piping
— reciprocating pumps- and compressors piping
— piping subject to slugging or flow induced vibrations
— gas piping with velocities larger than $V = 175 \times (1/\rho)^{0.43}$
— other services that typically can excite pipe vibration.

V - Velocity (m/s)

ρ - Density at operating condition (kg/m^3)

An alternative solution may be to reinforce the parent pipe in order to prevent fatigue failure due to shell vibration in parent pipe.

Branches having only minor weights attached do not require bracing, provided the branch is short enough to ensure adequate integral stiffness. When required, the branches shall be provided with bracings in two directions. Bracings shall preferably be made from L-profiles.

Bracing can be omitted if it can be demonstrated that the branch connections are not likely to be exposed to structural overload or vibration. For more information, see NORSOK L-002, 3rd edition.

3.17 Documentation of stress analysis

3.17.1 General

This section describes some minimum requirements to documentation of pipe stress analysis. The extent of documentation should be agreed upon between the owner, contractor and third party.

3.17.2 Documentation for the project and third party verification

Typical project deliveries from the pipe stress department should be:

— Pipe Stress Design Philosophy and/or Pipe Stress Work Instruction
— Pipe Stress Reports including typical appendices. Ref. 3.17.3 and 3.17.4 below.
— Electronic pipe stress FE-files (input files) that can be used to re-generate and run a pipe stress model by the company performing the 3rd party verification.

3.17.3 Requirement to a pipe stress report

3.17.3.1 The main sections of the pipe stress report

The main sections of the pipe stress report should contain relevant information given in the following typical sections within the stress report:

— Summary
— Introduction
— Scope of Work
— Regulations, Codes and Standards, Specifications
— Design Basis
— Design Philosophy, Design Considerations
— Load Cases
— Results
— Conclusion
— References
— Appendices.

3.17.3.2 Appendices

The pipe stress report should have typical appendices as listed below:

— Pipe Stress CAD Isometric index
— Pipe Stress CAD Isometrics, ref. 3.17.4 below. (May be issued as a separate document)
— 3D shaded plots of the pipe stress models
— Copy of the line list
— Pressure vessel-and equipment drawings with applied and allowable nozzle load tables
— Pipe Stress (FEA) input and output files. (Large volumes on enclosed CD or DVD)
— Flange stress- or flange leakage calculations
— PSV- and bursting disc relief load calculations
— Expansion and slip-joint thrust load calculations
— Flare tip ignition thrust load calculations
— Hydraulic hammer and surge load calculations
— Slug load calculations
— Pump, compressor and turbine combined nozzle load calculations
— Nozzle-to shell flexibility calculations
— Fatigue-and vortex shedding calculations
— Accidental load calculations
— Non standard piping component calculations
— Important vendor and project correspondence relevant to pipe stress analysis
— Any other applicable documentation for the stress approval of the piping system.

Some of the calculations listed above do not have to be documented in separate appendices if they are performed within the pipe stress analysis programme and listed in the output calculation report. Typical are flange calculations, nozzle flexibility calculations and pump, compressor and turbine combined nozzle load calculations.

3.17.4 Requirement to a pipe stress isometric

A pipe stress isometric shall have its piping layout extracted from the CAD system that is going to be used for the production of fabrication isometrics. So-called stress isometrics that can be printed from pipe stress programmes are not defined within this recommended practice to be a true pipe stress isometric. A typical pipe stress isomeric used by one of Europe's largest Piping Engineering Companies can be found in appendix F. (This has been enhanced in a CAD programme in order to remove unnecessary information for pipe stress analysis).

The isometrics that can be generated by pipe stress software will always by default show the correct dimensions and geometry according to the data that have been entered into the programme. Hence it is not possible by any 3rd part to trace any errors in routing and support positions if no CAD isometrics are available. It is important that there, as far as possible, is a 1:1 relationship between the Pipe Stress Model of the piping, and the CAD isometrics used during fabrication and installation of the piping.

For most current day projects there are client requirement stated in the project DFO (Documentation for Operation) relating to stress isometric drawings. One of these requirements is that scanned copies of marked up hard copies are not permitted. A stress isometric together with the stress engineer's comments should provide a complete summary of the analysis carried out with relevant output results. A stress isometric should be considered as a working document to be used by the operator and as such should meet a certain minimum standard required for stress isometrics.

A typical Pipe Stress Isometric should as a minimum contain the following elements:

a) The isometric has a paper size equal to A3 and it has a "Landscape-orientation".

b) Tables with relevant input data and results shall occupy no more than 1/3 of the total area of the sheet and the tables shall be placed on the right hand side of the sheet.

c) The icon describing the coordinate-axis should be placed in the upper or lower left corner and it is advisable to use the Z-axis as the vertical axis in order to have the same format as the structural department that is going to use the support loads for construction of pipe racks etc. It should be noted that most vendor equipment drawings uses the Y-axis as the vertical axis, and hence nozzle-loads must be carefully tabulated. (The latest revisions of most pipe stress programmes have the option to let the user select which axis that is going to be the vertical).

d) Information about line number, wall thickness, pipe spec, pipe material, fluid density, insulation, corrosion allowance, design-and operational temperatures and pressures, any design accelerations, wind speeds, design blast drag pressure, etc. shall be listed in tables at the upper right part of the Pipe Stress Isometric. This will be the input-data table used for modelling.

e) Below the input-data table, comes the summary table for highest calculated pipe stresses and the allowable stress limits for the actual case (sustain, displacement, operational, occasional, accidental) and a reference to the applicable piping code used for the analysis.

f) Below the table of calculated stresses comes the calculated pipe-support and nozzle load tables.

g) Below these tables comes the spring selection-and operational load setting table.

h) Below the spring selection tables comes the drawing information label. This will typically consist of the computer file name (run number), the drawing number, a logical name of the piping analysed (e.g. 14" HP Line from K.O. Drum to Flare") the actual revision of the sheet and fields for signatures, logos describing the Engineering Company, Operator, Project etc.

i) The left side of the sheet, where the CAD isometric has been copied and pasted into, shall show the routing of the piping, pipe-supports and their functions (refer Appendix F, *Pipe Stress Isometric* in this RP), unique identification of connected equipment nozzles and distances to the fixed point of this equipment.

j) Mark up of boundary conditions and interface piping such as branches that belong to a different pipe stress isometric/ analysis. This mark-up should be done with a dotted line to the first anchor support or equal function (two guides and a line stop, etc.) in order to have realistic boundary conditions.

k) Weights of valves and valve actuators and reaction forces from e.g. PSV valves shall be written close to the actual valves.

l) References to the relevant P & ID should be listed where it is space.

m) When there is not enough space for all important comments and calculation notes, these should be placed on an additional sheet. (A sheet with no isometric drawing, e.g. on page 2 of 2).

3.17.5 Documentation for Audit

Documentation and procedures that should be available during an audit of the project pipe stress department are:

— Pipe Stress Work Instruction and /or Pipe Stress Design Philosophy.
— Master files. That there is an updated master file for pipe stress isometrics where all essential changes that not yet have been handled are marked up.
— Data filing and storage. That computer FE analysis, "electronic hand-calculations" and project electronic communication are stored according to "pipe stress work instruction" and project QA specifications.
— A file cabinet etc with internally approved piping analysis consisting of: print-outs with "Yellow Line Check" of Input data and output results. Signed internal Check-Lists. (A typical example of an internal-or self-check list is given in Appendix B of this RP).
— Procedures for internal work-flow with the lay-out and design department, the pipe-support or structural department, process department etc.
— Procedures for coordination and communication with external companies, e.g. for pipe stress work carried out at a remote location or by sub-contractors. The issue is to verify that boundary conditions on interface piping designed at different locations are communicated between the locations in order to obtain a safe and functional design.
— CVs for the pipe stress engineers. The project pipe stress lead or other person responsible for the pipe stress analysis should be able to document by available CVs and dialog that he has the required education, training and practice as required by the ASME B31.3 Process Piping Code, chapter II, para. 301.1, *Qualification of the Designer*. "*Designer*" should in this context be replaced by "*Pipe Stress Engineer*".

3.18 Verification

Requirements to internal and external verification of offshore topside piping are given in section 6 of this recommended practice.

4. Subsea Piping

4.1 General

The intention with this section is to describe what a pipe stress engineer working with subsea piping systems normally has to consider with regard to pipe stress analysis. It is important to distinguish subsea piping systems from analysis of risers and pipelines even though some pipeline design codes are commonly used for analysis of subsea piping (e.g. ASME B31.4 and ASME B31.8).

4.2 Commonly used design codes

The following design codes have a long history in subsea applications and are commonly used in design of subsea piping systems:

- ASME B31.3 Process Piping Code
- ASME B31.4 Pipeline Transportation Systems for Liquid Hydrocarbons and Other Liquids
- ASME B31.8 Gas Transmission Distribution and Piping Systems.

ASME B31.4 (liquids) and ASME B31.8 (gas) are both a combination of typical pipeline and piping codes, but they have more in common with traditional pipeline codes than process piping codes.

4.2.1 ASME B31.3

The ASME B31.3 Process Piping Code is originally a design code for process plants to be placed on land. It is however the most used piping code for process piping on oil-and gas platforms and has been widely used for subsea installations. With the additional requirements to load cases and design calculations as listed in section 4.4 below, the additional design cases as listed in ASME B31.8 and API RP 17A (ISO-13628) are taken care of with respect to pipe stress analysis.

Subsea piping designed after ASME B31.3 will in general have a higher wall thickness than designed after the other codes. This may require a larger and heavier installation as the increased wall thickness requires more space (less flexible piping) to take the thermal-and imposed displacement loads. This again may require a larger installation barge.

4.2.2 ASME B31.4

This code is intended for transmission and distribution of liquids. ASME B31.4, section 400.1.2, clearly states that ASME B31.4 shall not be used for any subsea piping systems containing gas, such as hydrocarbons and CO_2.

4.2.3 ASME B31.8

This code is intended for transmission and distribution of gas in carbon steel pipelines. ASME B31.8, section 802.12, clearly states that the code shall not be used for any subsea piping systems containing liquids in any form or transportation of CO_2.

According to ASME B31.8, there is no need for derating of the material properties before a material temperature of 120°C. This temperature limit was ordinary based on that the code is intended for carbon steels, but today a lot of subsea installations are designed with piping in duplex materials and therefore the derating of the material properties should start at a lower temperature. (Refer DNV-OS-F101 for recommendations on derating temperature limits for other materials than carbon steels).

ASME B31.8 (gas) is more conservative than ASME B31.4 (liquids), e.g. with specific requirements to pipe supports being welded to the pipe, and probably the most used piping code for subsea installations. It is commonly used for other fluids and materials than allowed by the code. This is a dilemma for the designers and the 3rd party verificators, and hopefully the ASME committee will do something about it in the near future.

4.3 DNV-OS-F101 Submarine Pipeline Systems

This is a pipeline code developed by DNV. It will probably grow in popularity for design of subsea manifolds etc, and it has already been used in some Norwegian subsea projects in order to gain experience. It is based on the LRFD (Load and Resistance Factor Design) and not the ASD (Allowable Stress Design) methodology used for decades in traditional piping and pipeline design codes and also in the former DNV'96 pipeline code. An exception is however made for pipe bends that should be designed according to the ASD methodology. DNV-OS-F101 allows for much higher stress utilisation than any of the above mentioned design codes, and hence the wall thickness can be reduced which again increases the flexibility of the pipe. Increased flexibility again makes it possible to make a more compact design which is beneficial to the total weight of the installation as a smaller installation barge can be used.

A drawback with the code is that there is no commercial specialised pipe stress software available to day (2008) that includes this design code or the LRFD methodology in general. It is also doubtful that DNV-OS-F101 can be allowed for design of subsea piping in Europe that shall be qualified according to the Pressure Equipment Directive (PED), as DNV-OS-F101 allows for much higher stress levels than harmonised PED piping codes such as the EN-13480 piping code.

DNV-OF-F101 is ISO harmonised and contains information on HISC evaluations.

4.4 Recommended design cases

4.4.1 General

Subsea piping is normally designed according to the ASME B31.3 or the ASME B31.4/B31.8 Pipeline Codes. The DNV-OS-F101 Pipeline code has also been used for a few installations.

Below is a list of design cases which should be evaluated and analysed independent of design codes. The reason is that typical design codes such as ASME B31.3 is not intended for subsea or the offshore environment and hence does not list typical design cases found in pipeline codes such as ASME B31.8. (ASME B31.3 Piping design code leaves it to the designer to choose and analyse design cases that are not very common to process plants).

4.4.2 Recommended calculations and load cases for subsea installations

All calculations and design cases listed below should be considered in the design of subsea piping systems. All subsea piping shall be comprehensively analysed by use of computer FE programmes. The only exception is hydraulic control tubing that can be calculated by hand. Refer also Appendix H, Subsea Load Case Matrix, for typical combinations.

a) Hoop stress calculations for hydrostatic test pressure onshore and offshore.

b) Wall thickness and hoop stress calculations based on the pressure differential between internal and external (hydrostatic) design pressure. See actual design code for special requirements, e.g. ASME B31.3, ASME B31.4, ASME B31.8 or DNV-OS-F101.

c) Collapse- and buckling due to external hydrostatic over pressure. Ref. *ASME BPV, Div 1, Section VIII, UG-28 and DNV-OS-F101 Submarine Pipeline Systems, section 5.*

d) Lifting analysis onshore and offshore including dynamic loads and temporary piping displacement due to any sag of the subsea frame structure during lifting. Ref. *DNV Rules for Marine Operations, Part 2, Chapter 5, Lifting and Part 2, Chapter 6, Subsea Operations.*

e) Wave induced accelerations during transportation to the field.

f) Breaking waves, green sea, etc. hitting the piping during transportation to the field.

g) Lowering through the splash zone (wave slamming forces). Ref. *DNV Rules for Marine Operations Part 2, Chapter 6, Subsea Operations.*

h) Landing on seabed (additional g-forces).

i) Connection and Tie-in analysis (Pull-in).

j) Operational loads from connected pipelines.

k) Slug loads (and slug frequency to be used in any combined fatigue calculations).

l) Fluid hammering or surge forces due to valve and pump operations.

m) Earthquake.

n) Impact loads from iceberg-keel, trawl board, ROV, etc.

o) Dropped objects, e.g. anchors (piping to be protected).

p) Flange calculations (stresses and acceptable leak rate).

q) Local settlement of parts of the subsea skid frame that may cause twisting or bending of the subsea frame and thereby add bending moments to parts of the piping, valve and hub connections, etc.

r) Vortex shedding analysis (Ref. Appendix A).

s) Hydrogen Induced Stress Cracking calculations for relevant design cases (Ref. Section 4.5).

t) Acoustic fatigue evaluation for gas piping systems where duplex stainless steels are being used.

u) Fatigue calculations combining different sources such as VIV, slugs, pressure transients, etc.

v) Hot/cold system combinations, especially important for manifold piping.

w) Combinations of temperature, weight, pressure and imposed loads from attached pipelines, settlement, etc.

x) Any other design loads as listed for topside process piping where relevant.

It is recommended that piping transferring gas shall have full reinforcement plates 360 degrees around the pipe at location of pipe supports. (This requirement is taken from the ASME B.31.8 piping code). It is believed that this code requirement can have its origin in acoustic fatigue of duplex stainless steel welds for piping used to transport gas at high velocities. Acoustic high frequency sound may be generated inside long flexible risers when high velocity gas passes over the large number of carcass folds where vortices are generated).

4.5 Hydrogen Induced Stress Cracking (HISC)

4.5.1 Introduction

Both 22Cr and 25Cr duplex stainless steels have been extensively used for subsea piping and related piping components such as hubs and connectors. These types of steels have been used as pipes, castings and forgings. In general the experience is good, but some significant failures have occurred.

The main reason for these failures has been attributed to an unfortunate combination of load/stress and hydrogen embrittlement (HE) caused by ingress of hydrogen formed at the steel surface due to the cathodic protection. This is called Hydrogen Induced Stress Cracking (HISC).

Coating shall not be used as the only mean to prevent HISC by Cathodic Protection (CP). The combined material selection and design with respect to maximum allowable stress/strain shall be made such that HISC will not occur even if the coating is damaged or removed.

4.5.2 Recommended practice

Global stress analysis of subsea piping systems and local FE-analysis of special pressure containing components, such as connectors and hubs, should be designed with aim to avoid HISC. For more information on HISC, references are made to DNV-RP-F112 *Design of Duplex Stainless Steel Subsea Equipment Exposed to Cathodic Protection*.

4.5.3 Conflict with piping design codes

It is recommended to apply the guidelines set forth in DNV-RP-F112 in conjunction with the following designs codes: ASME B31.3, ASME B31.4, ASME B31.8 and ASME VIII Div. 2.

In case of conflict between the allowable design stresses and strains according to DNV-RP-F112 and a reference piping design code, the most stringent code should apply.

For those subsea piping installations where DNV-OS-F101 *Submarine Pipeline Systems* are being used instead of a traditional ASME code, the 2007 edition of DNV-OS-F101 now includes references to DNV-RP-F112 for HISC evaluation.

4.6 Documentation of pipe stress analysis

Documentation of subsea pipe stress analysis should in general be as for topside piping listed in section 3.17 but modified to the relevant design calculations special for subsea installations as mentioned in section 4.4.2 above. Small installations do no need to be documented with stress isometrics drawings as the one in Appendix F of this RP. Large and complex installations should however be documented by typical pipe stress isometrics as the example stress isometric given in Appendix F.

4.7 Verification

Requirements to internal and external verification of subsea piping should be as per section 6 of this recommended practice.

5. Non Standard Piping Components

5.1 General

Piping components such as bolts, fittings, valves, hubs, flanges, couplings, gaskets, etc. that are not designed according to a standard or specification accredited by the actual piping design code, should be qualified according to procedures outlined in pressure vessel codes referred to within the actual piping code. The same qualification is to be performed for standard piping components that are modified to suit a specific design or used outside its specified limits. There are however about 100 piping component standards referred to within ASME B31.3, so it is not often that piping engineers are involved in design of special components.

The main objective of this section is to describe the use of FEA to prove a component's resistance to excessive yielding (gross plastic deformation) and local failure (brittle fracture). In addition, some aspects related to proving the component's functionality and resistance to dynamic loading are covered.

The procedures described in this section do not apply when the material allowable stress at maximum design temperature is governed by the material's creep properties. (For standard carbon steels and for standard austenitic stainless steels, creep is not the governing factor for allowable stress before reaching a temperature of 400°C and +550°C respectively).

Components subjected to external pressure should be verified using applicable methods described in the piping design code or pressure vessel design code. Components that otherwise may fail due to buckling instability should be verified using relevant methods such as described in DNV-RP-C201 and DNV-RP-C202.

The methods described in this section are normally described as DBA; Design By Analysis.

Relevant design code references are:

a) ASME B31.3 Process Piping, 2006 Edition.
b) ASME B31.4 Pipeline Transportation Systems for Liquid Hydrocarbons and Other Liquids.
c) ASME B31.8 Gas Transmission Distribution and Piping Systems.
d) EN 13480:2002-3 Metallic industrial piping – Part 3: Design and calculation.
e) 2007 ASME Boiler & Pressure Vessel Code section VIII div.2. Alternative Rules. Rules for Construction of Pressure Vessels.
f) EN 13445-3:2002 Unfired pressure vessels – Part 3: Design.
g) ISO 13628-7:2005 Petroleum and natural gas industries – Design and operation of subsea production system – Part 7: Completion/workover riser systems.

5.2 Requirements in piping codes

Pressure equipment and piping components shall be documented to the requirements of the piping design code for the system they are a part of.

5.2.1 ASME B31.3 Process Piping code

Commonly used piping component standards that meet the code requirements are listed in Chapter IV, table 326.1 and Appendix A of the B31.3 code.

Piping components not listed in these parts of the code shall be designed and verified according to para. 302.2.3, 303 and 304.7.2 of the ASME B31.3 code.

In para. 304.7.2, reference is given to the ASME BPVC, Section VIII, Division 2, Appendix 4 and 6. These appendices give guidance and requirements to detailed stress analysis, e.g. by use of FEA and experimental stress analysis. In the re-written July 2007 issue of ASME VIII div.2, the applicable part is Part 5 Design By Analysis Requirements. The DBA methodology and acceptance criteria have been changed by implementing the LRFD (Load and Resistance Factor Design) method.

ASME B31.3 requires that the basic allowable stress from Table A-1 shall be used in place of allowable stresses in ASME BPVC Division 2. When applicable, the basic allowable stress shall be multiplied with the Casting Quality Factor, E_c see para. 302.3.3 and Table 302.3.3C.

5.2.2 EN Piping code EN-13480 Metallic industrial piping

EN13480 does not specifically say how non-standard components shall be documented. Components in such systems shall therefore be documented to a harmonized European standard, which is EN 13445 *Unfired Pressure Vessels.*

5.3 FEA design and qualification of non-standard piping components

The piping and pressure vessel design codes used for piping component design do not give much information on how FE analysis of piping components should be performed and documented.

Below is a description of acceptable methods for FE analysis and typical required documentation for third party verification by a Class Society, Notified Body, etc. A typical check list is included in Appendix D. It should be noted that in addition to FE analysis, subsequent post-processing of stresses shall be performed to show compliance with the design code. Such post-processing may be done by hand calculations based on output from the FE analysis or using code checking facilities in the FEA software.

The use of FEA to document the suitability of a component should be conducted by competent personnel with expertise both in the analysis method and in the results evaluation procedure.

If the DBA route is chosen, ref ASME VIII 2007 edition, div.2, Part 5, all of the stated design checks shall be considered.

5.3.1 FEA Methods to evaluate protection against plastic collapse

There are principally three main methods for verifying the plastic collapse strength of a component using FEA; elastic analysis, limit analysis based on elastic-perfectly plastic material model and small deformation theory, and plastic collapse analysis based on the actual strain hardening of the material and large deformation theory.

It is important to note that the linear elastic method involving stress linearization across sections is not always conservative. This applies in particular to heavy thickness components. Reference is made to ASME VIII div.2 part 5.2 and ISO 13628-7 Annex D for further discussion on the applicability of these methods.

The limit or plastic collapse methods may for many components be favourable over the linear elastic method since these methods do not involve the sometimes difficult and cumbersome work of stress categorisation, critical section identification and stress linearization.

5.3.1.1 Linear elastic FEA

The principle used in the pressure vessel design codes when verifying a component by linear elastic FEA is that critical sections shall be identified and verified by linearizing the stresses across the sections. Procedures for stress linearization are given in e.g. ASME VIII div. 2 Part 5. Several FEA programs include modules that perform stress linearization. In general, each of the six stress components shall be linearised separately, then calculate the principal stresses from these, for each of the stress categories (i.e. general and local membrane, bending and peak). The stresses shall further be categorized as either primary stresses or secondary stresses. The design codes have limits for allowable combinations of the linearised stresses for the two stress categories.

For components having pre-tension elements, stresses caused by the pre-tension force can be regarded as secondary stresses. However, the strength of the pre-tension elements shall be verified for all primary loads transferred though the connection.

For bolted flanges and similar connections, primary bolt stress shall be calculated from the total force in the bolt from internal pressure and other primary loads. The bolt stresses shall then be verified against the allowable stresses for primary membrane and membrane + bending stress for the actual type of loading.

Note that the design codes do not have a limit on maximum static secondary stresses. The limit on membrane + bending + secondary stresses of 3 times basic allowable stress which is found in several design codes, applies to the total stress intensity. For components where the loading giving maximum secondary stress in combination with primary stresses may affect the functionality of the component, the primary + secondary membrane stress shall be limited to 1.35 x basic allowable stress. This typically applies to connectors where the combination of primary and secondary loads may cause leakage. The evaluation of secondary membrane stresses is also applicable if the total membrane stresses are compressive and buckling instability is a possible failure mode. Relevant design checks shall then be performed.

For components where non-linearities such as contact behaviour is essential to include in the FE analysis, the method of code compliance check for linear elastic FEA may be used as long as the material is modelled as a linear elastic material. The code compliance check must then be performed at critical load steps in the non-linear analysis.

5.3.1.2 Limit analysis

In a limit analysis, it shall be verified that the actual loading is below the load that causes overall structural instability. The required safety against plastic collapse is ensured by applying a set of load factors to the different type of loads following an LRFD principle. The load factors may vary between design codes, but the total effect of the load factors for sustained loads such as pressure and dead weight, shall under no circumstance result in a lower factor of safety than 1.5 relative to yield for primary membrane stresses.

Note that when designing to the ASME VIII code, the yield stress to be used in the limit analysis shall be equal to 1.5 x basic allowable stress.

5.3.1.3 Plastic collapse analysis

In a plastic analysis, it shall be verified that the actual loading is below the load that causes overall structural instability. The required safety against plastic collapse is ensured by applying a set of load factors to the different type of loads following an LRFD principle. The load factors may vary between design codes and the load factors are generally higher than those used in limit analysis.

In plastic collapse analysis of assemblies, the realistic failure mechanism is simulated. Hence the required margin to failure shall be demonstrated for the load resisting properties of the assembly as a whole. This method will implicitly allow that bolts in a bolted connection can have a higher stress than basic allowable bolt stress since the bolt stress is due to a pre-stress giving the assembly as a whole the necessary load resistance.

5.3.2 FEA Methods to evaluate protection against local failure

Design codes normally have a simplified local stress check to be performed as a part of a linear elastic FEA. The stress check is based on limiting the sum of the principal stress components at any point in the structure (ASME codes) or limiting the maximum principal stress component at any point in the structure (EN codes).

Plastic collapse load analysis is well suited for verifying local failure since the actual structural behaviour is more closely approximated than in a limit analysis. It is required that local geometry is correctly described in the FE-model in order to determine local strains for subsequent code compliance check.

The local failure check is particularly important when using materials with very low ductility in combination with a geometry having local stress raisers such as small fillet radii or sharp corners. High strength grades of steel and titanium are in general more exposed to unstable fracture.

Maximum acceptable local stress can alternatively be determined by a fracture mechanics analysis, assuming the smallest surface defect detectable by non-destructive testing. Normally the fracture mechanics analysis will be of the type “fitness for purpose” based on Linear Elastic Fracture Mechanics (LEFM) with stresses taken from a linear elastic FEA.

For components in duplex stainless steels placed subsea and having a cathodic protection system, local strains are also necessary to be determined in order to demonstrate resistance to Hydrogen Induced Stress Cracking (HISC). Refer to section 4.5 of this RP for further information on HISC.

5.3.3 FEA Methods to evaluate protection against progressive collapse

Both ASME and EN design codes have ways for protection against progressive collapse from repeated loading. The limit analysis method should be used for this design check. The check for progressive collapse can be omitted if all loads are categorized as primary and at the same time meet the requirements for protection against plastic collapse.

5.3.4 FEA Methods to evaluate functionality

For components having complex geometry and loading, performing FEA is the best way of proving it’s functionality, apart from conducting actual tests. Normally, a non-linear analysis is required to properly check the functionality, e.g. check of contact surface separation that may cause leakage.

In a functionality check, the two extremities of material stiffness may often have to be evaluated. Only analysing the net thickness geometry as in the capacity analysis may be non-conservative when looking at functionality. E.g. contact stress may be over-predicted or assembly stiffness under-predicted. The greatest stiffness is represented by gross material thicknesses (un-corroded plus fabrication thickness tolerance) in combination with a linear elastic material. The smallest stiffness is represented by the net material thickness in combination with an elastic-perfectly plastic material.

When applicable, temperature effects in material properties should be accounted for.

For pre-loaded structures, sensitivity in pre-load should be evaluated based on the difference in thermal expansion between materials and any uncertainty in the pre-loading application procedure. Any long term relaxation effect should be accounted for.

5.3.5 The finite element model

The FE model of the component shall include necessary adjoining structures to assure that the application of loads and boundary conditions do not affect the stress and strain state in the critical sections. For components having a tubular end for welding to pipe (such as flanges, tees, weld-o-lets, etc.) a pipe length of minimum one diameter shall be included in the FE-model, measured from the location of the weld between component and pipe.

Axi-symmetric models may be used if the geometry is axi-symmetric and the loading is axi-symmetric. For components that are not truly axi-symmetric, but has one or more planes of symmetry, a symmetry model may be used as long as the loads and boundary conditions can be sufficiently defined on the model.

The modelling of bolt holes or other geometric details representing symmetrically positioned voids in the geometry must be done with outmost care in an axi-symmetric model. A reduced stiffness should be included in the model by appropriately reducing the Young’s modulus. Preferably, the method of reducing the stiffness for the type of component should be validated by a more detailed analysis using solid elements. This type of simplification in an axi-symmetric model should not be done in limit state or elastic plastic analyses.

Bending moments on pipes may be converted to an equivalent axial load F_{eq} enabling the use of a symmetry model.

$$F_{eq} = M_b \frac{32t(D-t)^2}{D^4 - (D-2t)^4}$$

For a bolted flange, the equivalent axial load F_{eq} shall be:

$$F_{eq} = \frac{4M_b}{BCD}$$

Where

D = pipe outer diameter
t = pipe wall thickness
M_b = resulting bending moment
BCD = bolt circle diameter

The element mesh and type of elements shall be suitable for the purpose of the analysis. Element size sensitivity analyses should be conducted to establish the required element density (or order of displacement function used for P-type FE programs) to describe the stresses sufficiently accurate to establish a reliable linearised stress field. (The P stands for Polynomial, where the polynomial order of the shape and displacement functions can be increased to much higher orders than those used in traditional H method analyses. Thus a P method model can analyse a problem with far fewer elements than would be needed by the H method to obtain the same accuracy).

Shell elements may be acceptable for components having a design that can be calculated by thin shell theory. Thick shells shall be used with great care in non-linear analysis.

Non-linear analysis shall be performed when applicable to account for effects like; geometric non-linearity (typically buckling), material non-linearity (e.g. yielding) and boundary nonlinearity (e.g. surface contact). For buckling instability analysis it is of vital importance that fabrication tolerances are included in the analysis. Such tolerances can e.g. be pipe out of roundness, and for welded parts; angular and alignment tolerances.

When creating FE models for the purpose of calculating stresses for subsequent fatigue analysis, care must be taken to ensure that the mesh density and level of detail modelled are in accordance with the assumptions in the chosen S-N curve. This is particularly important for welded components since S-N curves for welds include certain geometric effects. For more details see DNV-RP-C203.

For connections incorporating a seal between parts, the pressure shall be applied on the parts to the sealing diameter of the sealing element. For components having more than one sealing element in parallel, the sealing diameter of the outermost seal shall be used for pressure application.

5.3.6 Load combinations

In a linear elastic analysis, each type of load can be applied and run as separate load cases. In principle a unit load can be used, but in practice it is more convenient to apply a realistic value, e.g. the pressure load as the defined design pressure, the bending moment as maximum design bending moment, etc. The required load combinations by the applicable design code can then easily be performed during post-processing of the results by linear superposition of load cases.

Linear superposition of load cases cannot be performed in a non-linear analysis. When using the LRFD method in a limit or plastic collapse analysis, each required load combination shall be run as a separate analysis. The loads shall then be incrementally increased in load steps in which all loads are increased by the same factor. For pre-loaded components, the full pre-load shall be applied before the other loads.

If the ultimate load resistance is going to be determined by a limit or plastic collapse analysis, the load application sequence must be evaluated based on the anticipated load scenarios in operation. For pressure components being a part of a piping system having a certain pressure rating, it is recommended that the internal pressure including the end cap effect is applied first followed by load steps of external loads incrementally increased until the plastic capacity criteria is met, whether this is a strain limit or an instability limit.

5.3.7 FEA Reports

The FEA report shall as a minimum contain a description of the following:

a) Executive summary briefly describing the scope of the analysis and the main conclusion with reference to compliance with applicable design codes and DNV-OS.
b) Description of the component, its intended use with explanation of its functionality.
c) Reference to governing design specifications such as relevant DNV-OSS, DNV-OS and applicable pressure design codes.
d) References to project design premises and a summary of applicable loads and other design premises.
e) List of all relevant design drawings.
f) Component geometry analysed with reference to drawings. Any simplifications done in the geometry model should be discussed. This should include how fabrication tolerances and corrosion allowance are accounted for.
g) Materials including designation and reference material standard or specification. Lists of relevant material properties within the design temperature range.
h) The FE-model discretisation, with type of elements, discussion on element size with respect to accuracy in calculated stresses.
i) Description and colour plots of loads application and boundary conditions.
j) Results in the form of colour plots of stresses and strains. Plots of linearised stresses.
k) Code compliance check of stresses and strains for the limit states ULS, ALS and FLS including a clear conclusion with respect to code compliance.
l) Relevant functionality checks in the serviceability limit state.
m) Conclusion from FE model verification and load application verification including checking of reaction forces.

5.3.8 FEA check list

An example of a typical FEA checklist that should be used for local FE analysis of piping components is provided in Appendix D of this recommended practice.

6. Verification

6.1 General

This section gives recommended requirements to verification of pipe stress analysis.

6.2 Self check

A general check list is to be prepared by the company responsible for the pipe stress analysis and it shall be used for self-checking of own work. A typical check-list for topside piping is given in appendix B of this RP. A similar checklist should be made for subsea piping based on the special subsea stress calculations listed in section 4.

6.3 Internal verification

All pipe stress analysis work performed by a company is to have an internal verification performed by an experienced stress engineer. In projects where ASME B31.3 is used as the design code, either the person performing the analysis or the person that performs the internal verification of that persons pipe stress work, should fulfil the requirements to education and qualification as described in section 2.4.2.1 of this RP.

The stress isometrics and hence computer stress analysis should be considered to have been completed as checked when the checklist has been signed by both the analyst and the checker. Additional notes that pertain to the analysis, which add further relevance in support of the checking procedure should be appended to the checklist.

Checklists for all piping analysed shall be filed and be available to the customer or any 3rd party involved in audit or external verification.

6.4 Verification carried out by a 3rd part

6.4.1 General requirements

In general it should be the operator that has the responsibility to have the verification carried out. The verification should in principal not be delegated to the contractor who is responsible for the work that is to be verified.

An exception to this clause is piping designed for installations that are going to be placed in the European marked where the Pressure Equipment Directive, PED, is governing. Refer PED for requirements to verification, roles and use of Notified Bodies for verification of piping design.

There shall be organisational independence between those who carry out the design work, and those who verify it.

It shall be verified that provisions contained in relevant national and international regulations or decisions made pursuant to such regulations, have been complied with.

The extent of the verification and the verification method in the various phases shall be assessed. The consequences of any failure or defects that may occur during construction of the piping and its anticipated use shall receive particular attention in this assessment. The party carrying out the verification must be given the opportunity to carry out the verification in a satisfactory manner and time.

The verification shall confirm whether the piping satisfies the requirements for the specific location and method of operation, taking into consideration the design, including material selection and the analyses methods and programmes used.

Special consideration should be given to the organisation of verification activities in cases where new project execution models and/or information technology systems are introduced.

If an operator takes over a specification from another operator, verification may be omitted if this specification has previously been verified pursuant to the present regulations, and the specifications are otherwise applicable to the location in question and to the installation concerned.

6.4.2 Verification during the pre-engineering, detail engineering and follow on phase

Verification of design should include:

a) Specifications, etc. That project specification, selected design codes and standards, are in compliance with international and national regulations, Class rules, EU directives, etc.

b) Compliance with this recommended practice. That pipe stress analysis of topside or subsea piping as a minimum complies with this recommended practice.

c) Qualifications. That pipe stress personnel have the required qualifications.

d) Organisation. That the pipes stress department organises, and documents its work according to company QA procedures.

e) Software. The usefulness of computer software, and that the programmes are adequately tested and documented (especially Excel worksheets, MathCAD programs and similar). This is of particular importance when programmes are used in dealing with new problems, constructions or new software.

f) Deviations. That deviations during fabrication and installation are assessed and if necessary corrected.

g) Compliance. That calculations and FE models are in accordance with design drawings and piping and project specifications.

h) Non-standard components. That piping components not in compliance with any of the specifications and standards listed in ASME B31.3, Table 326.1 “Component Standards” are demonstrated to have sufficient mechanical integrity and documented according to the procedures outlined in section 5 of this RP.

i) Design review. That a design review is carried out by independent professional companies or consultants with personnel that, preferably, fulfil the minimum requirement to qualifications as listed in ASME B31.3, chapter II. A review should include the use of checklists such as the ones given in Appendix C and D of this RP.

j) The 3rd party should verify at least 5% of the total amount of the pipe stress models (piping systems) by independent stress analysis. Such analysis should preferably be performed by another pipe stress programme than used by the company being subjected to 3rd party verification. The remaining piping systems can be verified by receiving electronic pipe stress input files from the company being verified and run and analyse those pipe stress systems by the same software as used by the customer. Input data used in the customers pipe stress model must however be verified before the analysis is performed. The last option will be to just read the pipe stress reports and computer listing made by the company being verified and comment on these reports and conclusions. (The last option is however not recommended as the only method of verification).

6.4.3 Verification during the fabrication, installation and commissioning phase

The surveyor or person who verifies the pipework during these phases should check the following with pipe stress relevance:

a) Procedures. That satisfactory work instructions and procedures are prepared, e.g. procedures for alignment, welding, flange-bolt tightening, pressure- and leak testing.

b) Compliance. That the piping materials, lay-out, dimensions, insulation, valves, pipe supports, pipe-header, branches and by-passes comply with the pipe stress model or pipe stress isometric drawings.

c) Pipe support details. That pipe support limit-stops and required gaps as shown on pipe support detail-drawings, or as marked up on the pipe stress isometrics, have been physically installed and aligned with correct limit-stops/gaps.

d) Spring supports. That spring supports installed have the same spring rate as shown on the pipe stress isometric.

e) Lock pins. That spring support lock-pins are in place during lifting, transportation, hydro test, etc. and that these lock-pins have been removed from the spring-supports prior to commissioning, testing and operation.

f) Expansion joint bellows. That expansion joint type bellows without tie-rods have supports on either side with guides and line-stops that can take the pressure thrust load and that these line-stops are at the same locations as shown on the stress isometric drawings.

g) Alignment. That the alignment of pipe ends or pipe-flanges are done according to specifications prior to welding or bolt tightening.

h) Cold spring. That cold springing (forced bending, compression or stretching) of pipe spools are not carried out during installation in order to align flanges, nozzles, etc. This will introduce stresses in the pipe spools not considered for in the design and will be a possible hazard to other personnel in the future, e.g. during modification or maintenance work when the spools are being disconnected.

i) Blast supports. That specially designed blast supports that only shall have a function during a blast have been installed correctly. E.g. that required gaps are not closed etc.

j) Temporary supports. That temporary supports used for sea-fastening during transportation to the field or any temporary supports used during assembly of the piping have been removed before commissioning.

APPENDIX A
VORTEX INDUCED VIBRATIONS

A.1 General

Wind, current or any fluid flow past a structural component such as a cylinder may cause unsteady flow patterns due to vortex shedding.

Vortex induced vibrations (VIV) are related to elastic motion of an object such as a free-spanning pipeline.

The equations referred to in this appendix are taken from DNV-RP-C205 *Environmental Conditions and Environmental Loads*, section 9. The same equations are also listed in the older DNV Classification Note No. 30.5 *Environmental Conditions and Environmental Load.*

A.2 Relevance to piping

Subsea installations are commonly shielded by a structure to prevent dropped objects (e.g. anchors and heavy scrap) and over-trawling (fishing tool) from damaging the installation. Hence vortex shedding may not be a problem at all, but the density of the shielding may vary from project to project and the sea current velocity and possibility to pass through this shielding has to be checked out for the actual installation.

Project experience has also shown that the lowering speed of subsea units (installation phase) has been critical in deep water installations where it takes long time to reach the seabed and where the hydraulic-and the small bore piping are being exposed to oscillations from a vertical flow pattern.

For topside piping wind generated vibrations may show to be a problem for some piping in wind exposed areas such as the weather deck and the flare boom.

A.3 Scope

The scope of this appendix is to assist the piping stress engineers with guidelines on how to design topside and sub sea piping systems with guide-support spacing that will bring the natural frequencies of the piping systems outside the regions where in-line and cross-flow vortex shedding may lock onto the piping natural frequency and cause detrimental fatigue failure.

This appendix will only cover Vortex Induced Vibrations caused by wind (topside piping) and sea bottom current. VIV caused by wave particle velocity in shallow water are however of importance to riser and pipeline design. Refer DNV-RP-C205 for such calculations.

A.4 Important parameters

Important parameters governing vortex induced vibrations are:

— geometry (L/D)
— effective mass per unit length of pipe (m_e)
— added mass per unit length of pipe (m_a)
— damping ratio (ζ)
— reynolds number ($R_e=uD/\nu$)
— reduced velocity ($V_R=u/f_nD$).

where

L = pipe length between guides (m)
D = pipe diameter including insulation (m)
m_e = Effective mass per unit length (kg/m)
ζ = ratio between damping and critical damping
ρ = fluid density (kg/m^3)
ν = fluid kinematic viscosity (m^2/s)
u = flow velocity (m/s)
f_n = natural frequency of the pipe (Hz)

A.5 Vortex shedding frequency

For non-steady flow references are given to DNV-RP-C205, section 9.7, Wave Induced Vortex shedding.

The vortex shedding frequency in steady flow may be calculated as follows:

$$f_s = St\,\frac{u}{D}$$

where

f_s = vortex shedding frequency (Hz)
St = Strouhal number (= 0.2 for piping)
u = fluid velocity normal to the pipe axis (m/s)
D = pipe diameter including insulation (m)

Vortex shedding is related to the drag coefficient of the member considered. High drag coefficients usually accompany strong regular vortex shedding or vice versa.

For a smooth stationary cylinder, the Strouhal number is a function of Reynolds number (Re). The relationship between St and Re for a circular cylinder is given in Figure A-1 below.

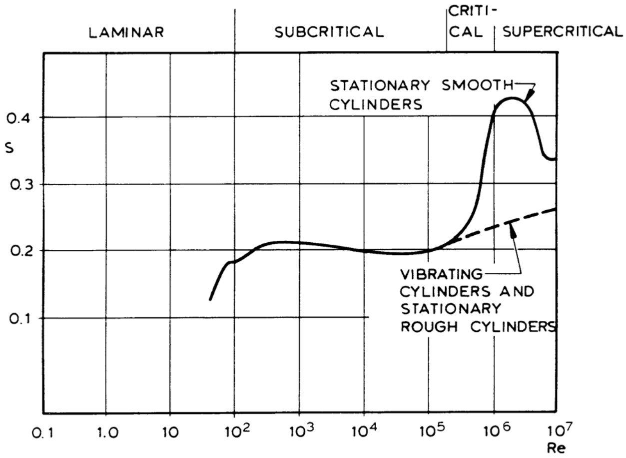


Figure A-1
Strouhal number, St, for a circular cylinder as a function of Reynolds number

A.6 Lock-in

At certain critical flow velocities, the vortex shedding frequency may coincide with a natural frequency of motion of the member, resulting in resonance vibrations.

When the flow velocity is increased or decreased so that the vortex shedding frequency *fs* approaches the natural frequency *fn*, the vortex shedding frequency locks onto the structure natural frequency and the resultant vibrations occur at or close to the natural frequency. This phenomenon is named “Lock-In” It should be noted that the Eigen frequency during lock-in may differ from the Eigen frequency in still water. This is due to variation in the added mass with flow velocity. It is beyond the scope of this RP to look at variations in added mass due to flow velocity, but a detailed description is given in DNV-RP-C205, section 9.1.12.

In the lock-in region, the vortex shedding frequency is dictated by the member’s Eigen frequency, while for lower and higher velocities the vortex shedding frequency follows the Strouhal relationship.

Lock-in to the piping Eigen frequencies can take place both as in-line and cross- flow VIV, described below.

A.7 In-Line VIV

In-line vortex induced vibration is defined as a vibration mode where the piping moves (vibrates) in a pattern parallel with the fluid flow, e.g. parallel with the sea bottom.

A.8 Cross-flow VIV

Cross-flow vortex induced vibration is defined as a vibration mode where the piping vibrates in a pattern perpendicular to the fluid flow, e.g. a free spanning pipeline parallel to the sea-bed that vibrates up-and down.

A.9 Reduced velocity, V_R

For determination of the velocity ranges where the vortex shedding will be in resonance with an Eigen frequency of the cylinder, a parameter V_R, called the reduced velocity, is used. V_R is defined as

$$V_R = \frac{u}{f_i D}$$

where

V_R	=	reduced velocity parameter
u	=	flow velocity normal to the member axis (m/s)
f_i	=	the i'th natural frequency of the member (Hz)
D	=	Pipe outer diameter including insulation (m)

A.10 Stability parameter, K_s

Another parameter controlling the motions is the stability parameter, K_s. It is also termed Scrouton number. This parameter is proportional to the damping and inversely proportional to the total exciting vortex shedding force. Hence the parameter is large when the damping is large or if the lock-in region on the member is small compared with the length of the pipe.

For uniform member diameter and uniform flow conditions over the member length the stability parameter is defined as

$$K_S = \frac{2\, m_e\, \delta}{\rho\, D^2}$$

where

K_s	=	Stability parameter
r	=	mass density of surrounding medium (kg/m^3)
D	=	Pipe diameter including insulation (m)
m_e	=	effective mass per unit length (kg/m)
δ	=	the logarithmic decrement ($=2\pi\zeta$)
ζ	=	the ratio between damping and critical damping. (If no data are available then $\zeta = 0.005$ if submerged in seawater and $\zeta = 0.0015$ for air can be used).

A.11 Effective mass, m_e

The effective mass (m_e) per unit length of the pipe consists of the following weight elements:

- the pipe it self
- insulation
- internal content
- added mass, m_a (se equation below).

When calculating the effective mass, no gain shall be taken for the buoyancy effect of submerged piping.

A.12 Added mass, m_a

The added mass is in general the mass of the volume of the fluid which is displaced by the pipe per unit length.

For simplified VIV analysis of a straight pipe (without valves etc) the added mass can be calculated as

$$m_a = C_a \cdot (\rho \cdot \frac{\pi}{4} D^2)$$

where:

m_a	=	added mass per unit length (kg/m)
C_a	=	added mass coefficient (use 1.0 for piping)
ρ	=	mass density of surrounding medium (kg/m^3)
D	=	Pipe diameter including insulation (m)

A.13 Limitation to added-mass equation

Above equations can not be used for hand calculations of sub-sea piping with free-hanging and unsupported valves-and valve actuators. Use FE analysis for such calculations. The value of the added mass coefficient may also depend on the boundary conditions. For more information reference is made to the literature listed in DNV-RP-C205, section 9.

A.14 Common error in pipe stress software

Note that several of the most commonly used pipe stress programmes for subsea and topside piping per 2008 do not include the added mass from seawater (or air) in the effective mass per unit length. Hence, to correct for this error, it will be necessary to increase the density of the pipe material, insulation or content in order to compensate for the missing weight. Use hand calculations to verify that the total mass per pipe unit length calculated by the pipe stress programme is correct. Any default pipe buoyancy calculation should also be turned off during this manual correction to obtain correct added mass.

If the added mass from surrounded seawater is not taken care of by manipulation of the pipe or content densities, the modal analysis of submerged piping will show wrong natural frequencies.

A.15 Wind induced vortex shedding

A.15.1 General

Wind induced cyclic excitations of pipes may occur in two planes, in-line with or perpendicular to (cross flow) the wind direction.

Vortex shedding induced oscillations due to wind are similar to the vortex shedding in steady current. It should be noted that the mass ratio for wind-exposed structures are normally much larger than for structures in water. The high mass ratio tends to give a smaller lock-in area and the time required to build up resonance oscillations increases. This means that the onset criteria are different for wind and current flows, while the amplitudes as a function of K_s for fully developed vortex induced oscillations are similar.

A.15.2 In-line oscillations

In-line excitations in air may occur when:

$$1.7 < V_R < 3.2$$

In-line oscillations are not likely to occur unless there are large concentrated masses excited, e.g. valves.

A.15.3 Cross flow oscillations

Cross flow excitations in air may occur when:

$$0.85 \frac{1}{St} < V_R < 1.6 \frac{1}{St}$$

The Strouhal number for circular cross sections such as piping may be taken as St = 0.2.

A.16 Sea current induced vortex shedding

A.16.1 In-line oscillations

In-line VIV is separated into pure in-line and cross flow induced in-line motion. The latter will not be discussed here, and hence references are given to DNV-RP-C205, section 9.

Pure in-line vortex shedding resonance (lock-in) may occur for submerged piping when:

$$1.0 \leq V_R \leq 4.5$$

$$K_S \leq 1.8$$

Depending on the flow velocity, the vortices will either be shed symmetrically or alternatively from either side of the cylinder.

For $1.0 < V_R < 2.2$, in the *first instability region*, the shedding will be symmetrical. The onset criterion is only valid when the reduced velocity V_R is increasing. In non-steady flow where V_R may go from high values to low values, lock-in vibrations will exist for all $V_R \geq 1.0$.

For $V_R > 2.2$ the shedding will be asymmetrical and the motion will take place in the *second instability* region ($2.2 < V_R < 4.5$) for $K_S < 1.8$.

A.16.2 Cross flow oscillations

Cross flow vortex shedding excitation may occur for submerged piping when:

$$3 \leq V_R \leq 16$$

A.17 Recommended VIV check-out procedure

A.17.1 General

Below is a recommended simplified design check for piping that might be exposed to vortex induced vibrations from wind (topside piping) or sea currents (subsea piping).

Pipelines, Risers and wave-induced particle velocities at sea bottom for shallow water installations are not covered by this appendix. Refer to DNV-RP-C205 *Environmental Conditions and Environmental Loads* for information about wave induced particle velocities and VIV related to that.

A.17.2 VIV Check-out procedure

a) Calculate the natural frequencies of the piping, preferably by FE modal analysis. Be aware that most widely sold and commonly used pipe stress FE programmes per 2007 by mistake ignore the added weight from surrounding seawater. Refer to section A.11-A.14 for necessary corrections. For straight pipes without free-hanging valves, etc. the following equation can be used for hand calculations of the piping natural frequencies:

$$f_n = \frac{a}{2\pi L^2}\sqrt{\frac{EI}{m_e}}$$

where:

f_n = Natural frequency of the pipe (Hz)
a = fixation constant depending on boundary conditions and mode shapes
L = Span length between pipe support guides or fixation support (m)
E = Young's Modulus for pipe material (N/m^2)
I = Moment of Inertia (m^4)
m_e = Effective mass per unit length of pipe (kg/m)

Table A-1 Values for the fixation constant, a, depending on boundary conditions and mode shape

Boundary condition	*1st Mode*	*2nd Mode*	*3rd Mode*
Guide-Guide	a = 9.87	a = 39.5	a = 88.9
Guide-Fixed	a = 15.4	a = 50.0	a = 104
Fixed-Fixed	a = 22.4	a = 61.7	a = 121

b) Calculate the reduced velocity, V_R, as described for wind (A.15) or sea currents (A.16) above.

c) For wind-exposed piping, check that the reduced velocity, V_R, is outside the regions for which In-line and Cross-flow Vortex Induced Vibrations may occur. (A.15.2 and A.15.3).

d) For subsea piping, calculate the stability parameter *Ks*, according to the equation in A.10 above and check the validity of the equation for In-line VIV (check that $K_S \leq 1.8$, ref. A.16.1). Then check that the reduced velocity, V_R, is outside the regions for which In-line and Cross-flow Vortex Induced Vibrations may occur. (A.16.1 and A.16.2).

e) If the calculations show that vortex induced vibrations are likely to be expected, try out the methods for reducing the vibrations as described in A.18 below. If still a problem, or if it is too late in the project design phase to change the layout, perform a detailed fatigue analysis taking all relevant fatigue sources into considerations. (Both high cycle and low cycle fatigue sources). Such sources may be temperature and pressure transients, slugs, fluid-hammer, surge, acoustic caused vibrations in high velocity gas piping etc. It is recommended to use the method outlined in PD5500 for such combined fatigue calculations.

A.18 Methods used to reduce VIV

A.18.1 General

If calculations (including fatigue) have shown that it is not possible to live with the vortex-induced oscillations, there exist a couple of ways for reducing them.

These methods are:

— changing the layout and support-spacing
— shielding.

A.18.2 Changing the layout and spacing between pipe supports

For topside (e.g. flare tower piping) and subsea installations the only practical way to avoid vortex-induced oscillations is to manipulate the natural frequencies by adding or changing the spacing between the pipe support guides. For subsea templates a different routing of the pipe, e.g. parallel instead of perpendicular to the main current direction, may be a solution.

A.18.3 Shielding

If it is not possible to change the natural frequencies by adjusting the spacing of the guides (support) it may be possible to shield minor piping from the main wind directions in front of, or behind, larger steelwork or piping. Typical are small bore flare-tip ignition (pilot) lines and piping along a bridge between two platforms.

Subsea installations are commonly shielded by a steel or composite structure in order to prevent dropped objects (e.g. anchors and heavy metal scrap) and over-trawling (fishing tool) from damaging the installation. Hence vortex shedding may not be a problem at all, but the density of the shielding may vary from project to project and the current velocity and possibility to pass through this shielding has to be checked out with the layout and project design basis.

A.19 Physical properties of air and seawater

A.19.1 Densities and cinematic viscosities

The below values for fluid density and cinematic viscosities are guidance values to be used for dry air and seawater. Cinematic viscosity parameter is used to calculate the Reynolds number in order to find the Strouhal number. An average value of the Strouhhal number for piping can however be set to St = 0.2 (Refer Figure A-1, above).

Table A-2 Density and cinematic viscosity for dry air and seawater

Temp [°C]	*Density, ρ, [kg/m³]*		*Kin. Visc, ν, [m²/s]*	
	Sea Water (1)	*Dry Air (2)*	*Sea Water (1)*	*Dry Air*
0	1028	1.293	1.83 E-6	1.32 E-5
5	1028	1.270	1.56 E-6	1.36 E-5
10	1027	1.247	1.35 E-6	1.41 E-5
15	1026	1.226	1.19 E-6	1.45 E-5
20	1025	1.205	1.05 E-6	1.50 E-5
25	1023	1.184	0.94 E-6	1.55 E-5
30	1021	1.165	0.85 E-6	1.60 E-5

Notes:

1) Salinity of seawater = 3.5%
2) Density at atmospheric conditions, Pa= 1013 kPa

APPENDIX B
ANALYST CHECK LIST, GLOBAL ANALYSIS

<table>
<tr><th colspan="5">ANALYST CHECK LIST, GLOBAL ANALYSIS</th></tr>
<tr><td colspan="2">STRESS CALC. NUMBER:</td><td></td><td colspan="2" rowspan="2">Revision:</td></tr>
<tr><td colspan="2">SYSTEM DESCRIPTION:</td><td></td></tr>
<tr><td>ID</td><td>REFERENCE</td><td>DESCRIPTION</td><td>Used</td><td>Chk'd</td></tr>
<tr><td rowspan="24">COMPUTER INPUT DATA</td><td>Piping Code</td><td>Is correct piping code (B31.3, B31.4, B31.8, EN13480 etc) used?</td><td></td><td></td></tr>
<tr><td>Numbering</td><td>Is the calculation number correct and according to project procedure?</td><td></td><td></td></tr>
<tr><td>Units</td><td>Are correct units used?</td><td></td><td></td></tr>
<tr><td>Pipe Spec</td><td>Is the correct pipe spec used (diameter, thickness, material properties, corrosion allowance, fabrication tolerances, weld joint factor etc)?</td><td></td><td></td></tr>
<tr><td>Fluid</td><td>Is the density of internal fluid correct or conservative?</td><td></td><td></td></tr>
<tr><td>Pressure</td><td>Correct pressures (design, operating, hydro test)?</td><td></td><td></td></tr>
<tr><td>Temperature</td><td>Correct temp (installation, ambient, operating, high & low design)?</td><td></td><td></td></tr>
<tr><td>Insulation</td><td>Insulation density and thickness?</td><td></td><td></td></tr>
<tr><td>Weights</td><td>Correct weight of flanges, bolts, valves and valve actuators?</td><td></td><td></td></tr>
<tr><td rowspan="3">Equipment</td><td>Are the locations of fixed and sliding supports correctly modelled?</td><td></td><td></td></tr>
<tr><td>Are nozzle movements due to temperature, live-and dead loads applied?</td><td></td><td></td></tr>
<tr><td>Is nozzle to shell flexibility correctly applied, if used?</td><td></td><td></td></tr>
<tr><td>Acceleration</td><td>Earthquake, Wave-induced, Lifting, Landing, directions</td><td></td><td></td></tr>
<tr><td>Deflection</td><td>Structural Deflections (Live load, Lifting, SAG/HOG etc)</td><td></td><td></td></tr>
<tr><td>Blast</td><td>Is the drag pressure, structural deflections, Cd, and DLF correct?</td><td></td><td></td></tr>
<tr><td>Wind</td><td>Is the wind speed, relevant directions and Cd coefficient correct?</td><td></td><td></td></tr>
<tr><td>Reaction Forces</td><td>Typical: PSV relief, Bursting Disc, Flare Ignition, Fluid Hammer, Surge, Slugs, adequate use of dynamic load factors, DLF</td><td></td><td></td></tr>
<tr><td>Dynamic Analysis</td><td>Unrealistic results are caused by springs, gaps, friction, no guides, no added mass from seawater, etc.</td><td></td><td></td></tr>
<tr><td>Load cases</td><td>Relevant combination of load cases including code req.</td><td></td><td></td></tr>
<tr><td>3D Model Geometry</td><td>Coordinates and pipe routing according to ISO drawing?</td><td></td><td></td></tr>
<tr><td rowspan="2">Restraints/supports</td><td>Are functions according to stress isometrics or pipe support drawing?</td><td></td><td></td></tr>
<tr><td>Are the restraints modelled realistically (gaps, stiffness, friction, etc.)?</td><td></td><td></td></tr>
<tr><td>Fatigue</td><td>Are fatigue cycles and stress range input realistic (ref. PD 5500)?</td><td></td><td></td></tr>
<tr><td colspan="4"></td></tr>
<tr><td rowspan="8">CALCULATIONS AND OUTPUT</td><td rowspan="4">FEA Calculation output</td><td>Have all relevant load-cases been analysed? Are combinations of load-cases realistic?</td><td></td><td></td></tr>
<tr><td>Are deflections and loads at restraints and equipment nozzles OK?</td><td></td><td></td></tr>
<tr><td>Are strains and stresses calculated within code allowed and is the allowable stress correct according to the piping code used? Have SIFs and SCFs been included according to code requirements?</td><td></td><td></td></tr>
<tr><td>Are results from fatigue and dynamic analysis realistic and conservative?</td><td></td><td></td></tr>
<tr><td>Stress Iso Drawing</td><td>Check tables with input data & output results. Data nodes, pipe-supports, weights, boundary conditions, references and notes. Signed the drawing.</td><td></td><td></td></tr>
<tr><td rowspan="3">Hand calculations to be included in the Pipe Stress Report Appendices</td><td>Have thrust-load calculations used as input for FEA been reviewed and filed?</td><td></td><td></td></tr>
<tr><td>Have stress calculations of special components, flange leakage calculations and fatigue calculations been performed?</td><td></td><td></td></tr>
<tr><td>Special vendor info such as reaction loads from PSV and flare tip filed?</td><td></td><td></td></tr>
<tr><td colspan="2">Use of Checklist:</td><td colspan="3">Y=Yes, N=No, OK=OK NA= Not Applicable, NC= Not Checked</td></tr>
<tr><td colspan="3">Analysis by:</td><td colspan="2">Date:</td></tr>
<tr><td colspan="3">Checked by:</td><td colspan="2">Date:</td></tr>
</table>

APPENDIX C
THIRD PARTY CHECK LIST, GLOBAL ANALYSIS

THIRD PARTY CHECK LIST, GLOBAL ANALYSIS

This checklist should be used by any 3rd party pointed out by the company being reviewed, Notified Body, Class Society, Contractor or National Authorities.

PROJECT:		Revision:
PIPE STRESS REPORT No:		

ID	REFERENCE	DESCRIPTION	Used	Chk'd
PROJECT SPECIFICATIONS	International regulations	Are international regulations complied with, e.g. the PED directive?		
	National regulations	Are National regulations complied with (UK, Canadian, Norwegian, etc.)?		
	Class Society	Are correct Class Society rules complied with (ABS, DNV, Lloyds, etc.)?		
	Piping Code	Is the correct piping code used (ASME B31.3, B31.4, B31.8 EN 13480)?		
	Environmental loads	Are environmental design data in accordance with project specifications?		
	Accidental loads	Are accidental design data in accordance with project specifications?		
	Operator Specifications	Are any special requirements from the operator complied with?		
	Company Specifications	Are any Company Specifications approved by the Operator complied with?		
FE- ANALYSIS AND HAND CALCULATIONS	Topside piping	Have all load cases and calculations at least been performed (or discussed) according to equations and methodologies given in this RP or in the relevant codes, standards, and recommended practices as listed in the REFERENCE and TOPSIDE sections of this RP?		
	Subsea piping	Have all load cases and calculations at least been performed (or discussed) according to equations and methodologies given in this RP or in the relevant codes, standards, and recommended practices as listed in the REFERENCE and SUBSEA sections of this RP?		
	Input data	For each Pipe Stress Analysis you review, use the "ANALYSTS CHECK-LIST" in this RP (previous page) and check off for each check point listed in the "Computer FEA Input Data" section what you have checked. Have you performed this check for the actual analysis?		
	Calculations and report	For each Pipe Stress Analysis you review, use the "ANALYSTS CHECK-LIST" in this RP (previous page) and check off for each check point listed in the "CALCULATION OUTPUT" section what you have checked. Have you performed this check for the actual analysis?		
	Independent analysis	Has this system been subjected to independent 3rd party stress analysis?		
DOCUMENTATION AND QA	Qualifications, CV	Request the CV of the person responsible for the Pipe Stress Analysis. Refer to this RP or ASME B31.3, chapter II. Is he qualified according to the strict requirements to education, training and practice as listed in ASME B31.3?		
	Pipe Stress Report	Typical main sections are: Summary, Introduction, Scope of Work, Regulations, Codes and Standards, Specifications, Design Basis, Design Conditions, Design Load Cases, Results, Conclusion, References and Appendices. Does the main report contain these typical main sections?		
	Pipe Stress Report-Appendices	Typical appendices consists of: Pipe Stress Isometric Index, Pipe Stress Isometrics Drawings, 3D shaded model plots, line-list, Pressure vessel drawings with nozzle-load tables, listing of Computer FE-Analysis, Flange calculations, PSV- and Bursting Disc Relief Load calculations, Slug calculations, Flare tip thrust load calculations, Pump-and Compressor/Turbine load calculations, Nozzle-shell flexibility calculations, Hydraulic hammer and surge calculations, Fatigue-and Vortex Shedding Calculations, HISC calculations, important vendor information, important correspondence, etc . Do the appendices contain this information?		
	Electronic 3D files/computer input file	Have electronic Pipe Stress Computer Models/Input Files been copied on a CD/DVD and attached to the stress report?		
	Filing	Has the report, analysis and checklist been properly filed?		
Use of Checklist:		Y=Yes, N=No, OK=OK NA= Not Applicable, NC= Not Checked		

Third party review by:	Date:
Checked by:	Date:

APPENDIX D
CHECK LIST, LOCAL FE-ANALYSIS

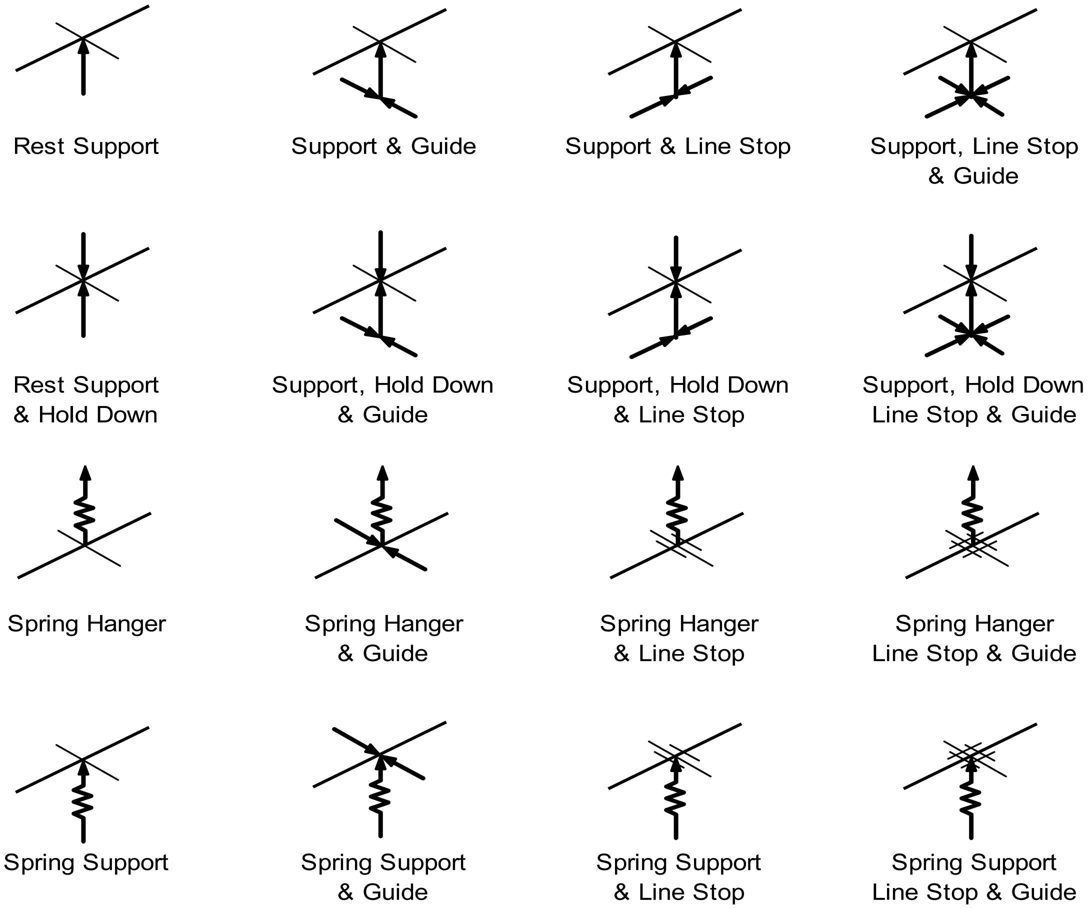
CHECK LIST, LOCAL FE-ANALYSIS OF PIPING COMPONENTS					
PROJECT:				Revision:	
ITEM DESCRIPTION:					
ID	REFERENCE	DESCRIPTION		Used	Chk'd
PROJECT SPECIFICATIONS	International regulations	Are international regulations complied with, e.g. the PED directive?			
	National regulations	Are National regulations complied with (UK, Canadian, Norwegian, etc.)?			
	Class Society	Are Class Society rules complied with (ABS, DNV, Lloyds, etc.)?			
	Piping Code	Are the correct piping code and corresponding pressure vessel code used for the design of this component e.g. ASME B31.3 & ASME VIII Div2 part 4/5?			
	Drawings, materials, design basis	Are all component drawings, material data, project design basis, etc. received and complied with?			
FE- ANALYSIS AND HAND CALCULATIONS	3D-Model	Is the geometric 3D model of the piping component in accordance with project specifications? Does it have sufficient level of details? If used, are symmetry planes selected correctly? Is the centre of gravity and weight correct? Does the model contain unwanted gaps?(A third party may need access to the original or a universal (neutral) file-format of the 3D CAD file in order to be able to verify the model against drawings).			
	Material properties	Are the material properties at the operating and design temperatures correct?			
	Type of Analysis	Is the chosen FE-analysis appropriate for the loading conditions (linear static analysis, nonlinear static analysis, contacts analysis, etc.)?			
	Restraints, loads and other boundary conditions	Have the restraining, loads, pre-described displacements, contact surfaces and boundary conditions in general been applied correctly? Have colour plots clearly showing these topics been included in the report?			
	Element type	Is the element type (beam, truss, shell, solid, etc.) used for the FE analysis appropriate for the part(s) being analysed? If solid elements are used, is the chosen element suitable for the part under analysis?			
	Meshing	Is the mesh density (element size) confirmed accurately by sensitivity analysis or other suitable method? Are stress precision values and/or mesh convergence studies OK? Has the mesh been checked and found OK in critical areas?			
	Results	Are the stresses, strains, rotations, deflections, etc. within the design code allowed or project specifications? Is penetration of contact surfaces avoided? Has the methodology outlined in the governing Pressure vessel code (e.g. ASME VIII, Div2, part 4 or 5) been complied with?			
	Hand calculations	Have some basic hand calculations been performed to roughly estimate and validate the FE results? E.g. are the reaction forces from the FE-analysis checked against applied loads?			
	Independent Analysis (Third party verification)	Has any third party FE- analysis been performed to validate the FE-analysis of the component?			
DOCUMENTATION AND QA	Qualifications, CV	Is the person responsible for this FE analysis considered competent in his field and does the person have sufficient analytical skills, experience or back-up to make a safe design in accordance with project specifications? problems?			
	FEA Report	Does the FEA report as a minimum contain the information listed in section 5.3.7 "FEA Report" of this recommended practice?			
	Electronic 3D CAD files	Have the original or some electronic universal (neutral) file-format for the 3D CAD or FE model been made available to the project for third party review?			
	Filing	Has the report, analysis and checklist been properly filed?			
Use of Checklist:		Y=Yes, N=No, OK=OK NA= Not Applicable, NC= Not Checked			
Checked by/Third party review by:			Date:		

APPENDIX E
RESTRAINT SYMBOLS

The following restraint symbols should be used on the pipe stress isometric. Where uncertainty may exist in the use of one symbol or another, a simple description should accompany the symbolic representation of the supports.

Typical descriptions are: RS = Rest Support, HD =Hold Down, LG =Line Guide, LS = Line Stop.

Restraint description for a coordinate system with the +Y axis pointing upwards could then be: LGX = Line Guide in ± X direction, LGZ = Line Guide in ± Z direction, LSX = Line Stop in ± X direction, LSZ = Line Stop in ± Z direction.



It is acceptable to use the arrow-symbols for line-stops and guides in combination with spring-supports.

Gaps or limit stops:

In cases where it is necessary to have gaps or limit stops at guides and line-stops, this additional information shall be written in text close to the actual restraint. Example:
"Guide w/Gap = ± 10 mm". "Line-stop w/Gap = ± 5 mm".

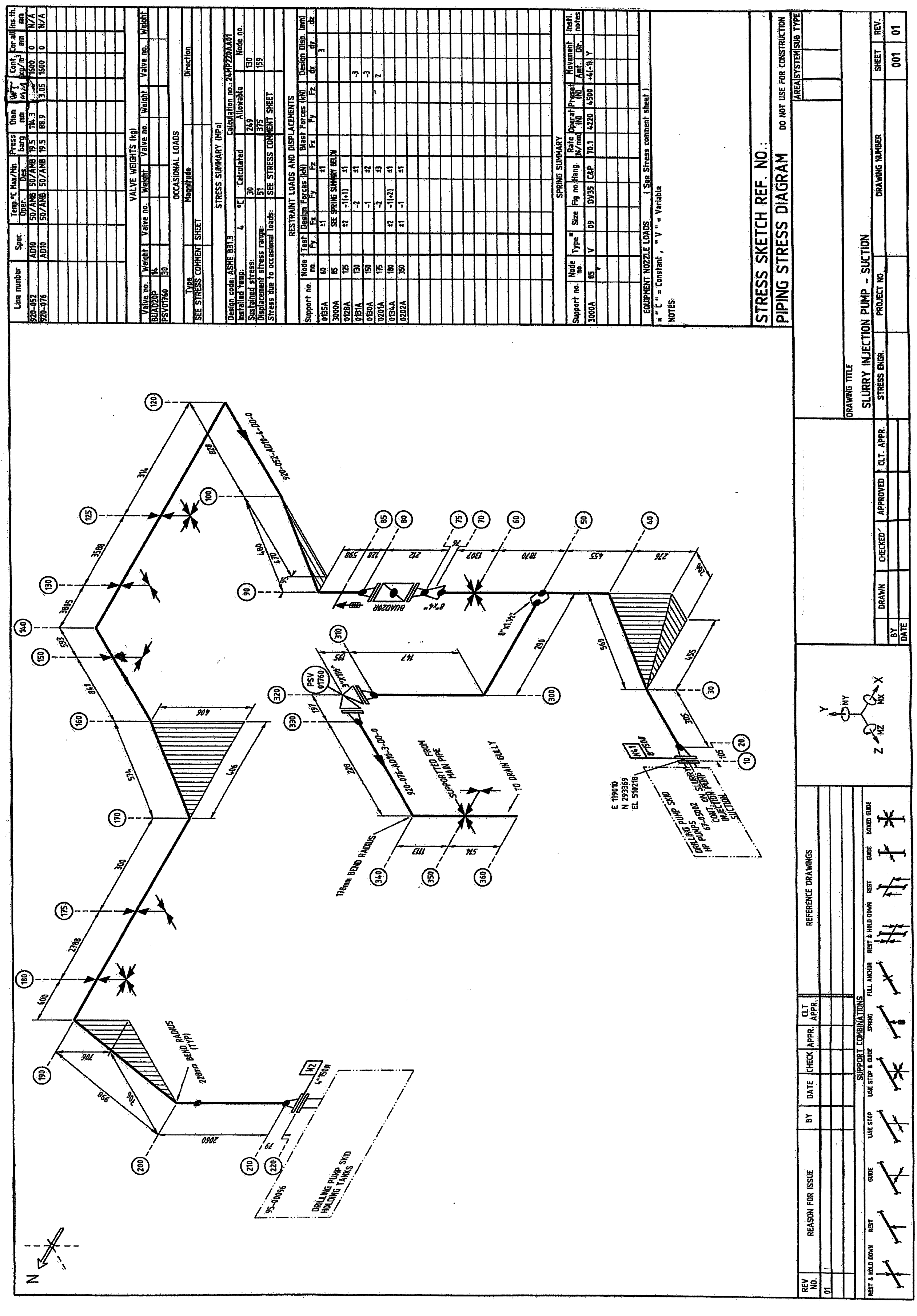
Note:

The most commonly (by default) used coordinate system for pipe stress analysis follows the "right hand rule" with the +Y axis pointing upwards. For compliance with FEA programs used for structural steelwork calculations and reporting of pipe support loads to the structural department, it may be beneficial to choose the Z axis to be the vertical axis.

---e-n-d---of---N-o-t-e---

APPENDIX F
PIPE STRESS ISOMETRIC

Refer also section 3.17.4 'Requirement to a pipe stress isometric'.



APPENDIX G
LOAD CASE COMBINATIONS

Typical project load case combinations for operatinal and occasional loads. (Accidental blast loads and other accidental loads have to be treated separately).

Case No.	*Load Case Combination Design Runs*	*Description*	*Stress Category*	*Combine Method*	Output
1	W+D1+T1+P1+H	Maximum Design Conditions 1	(OPE)	-	Disp/Force
2	W+D2+T2+P1+H	Minimum Design Conditions 2	(OPE)	-	Disp/Force
3	W+D3+T3+P1+H	Normal Operating Conditions	(OPE)	-	Disp/Force
4	W+P1+H	Weight + Design Pressure + Spring Force	(SUS)	-	Disp/Force/Stress
5	WW+HP	Hydrotest Pressure	(SUS)	-	Disp/Force/Stress
6	W+D1+T1+P1+H+U1	Maximum Design Conditions 1+U1	(OPE)	-	Disp/Force
7	W+D1+T1+P1+H+U2	Maximum Design Conditions 1+U2	(OPE)	-	Disp/Force
8	W+D1+T1+P1+H+U3	Maximum Design Conditions 1+U3	(OPE)	-	Disp/Force
9	W+D1+T1+P1+H+WIN1	Maximum Design Conditions 1+WIN1	(OPE)	-	Disp/Force
10	W+D1+T1+P1+H+WIN2	Maximum Design Conditions 1+WIN2	(OPE)	-	Disp/Force
11	W+D1+T1+P1+H+F1	Maximum Design Conditions 1+F1	(OPE)	-	Disp/Force
12	L6-L1	Acceleration Vector 1 (only)	(OCC)	Algebraic	Disp/Force/Stress
13	L7-L1	Acceleration Vector 2 (only)	(OCC)	Algebraic	Disp/Force/Stress
14	L8-L1	Acceleration Vector 3 (only)	(OCC)	Algebraic	Disp/Force/Stress
15	L9-L1	Wind North (only)	(OCC)	Algebraic	Disp/Force/Stress
16	L10-L1	Wind West (only)	(OCC)	Algebraic	Disp/Force/Stress
17	L11-L1	Relief Valve Reaction Load (Only)	(OCC)	Algebraic	Disp/Force/Stress
18	L1-L4	Thermal 1 + Disp 1 (Max Design)	(EXP)	Algebraic	Disp/Force/Stress
19	L2-L4	Thermal 2 + Disp 2 (Min Design)	(EXP)	Algebraic	Disp/Force/Stress
20	L3-L4	Thermal 3 + Disp 3 (Normal Operating)	(EXP)	Algebraic	Disp/Force/Stress
21	L1-L2	Displacement Stress Range T1-T2	(EXP)	Algebraic	Disp/Force/Stress
22	L12+L13+L14	Resultant Acceleration Vector	(OCC)	SRSS	Disp/Force/Stress
23	L15+L16	Resultant Wind	(OCC)	SRSS	Disp/Force/Stress
24	L4+L17	Weight + Pressure + Relief Valve Reaction	(OCC)	Scalar	Disp/Force/Stress
25	L4+L22	Sustained + Accelerations	(OCC)	Scalar	Disp/Force/Stress
26	L22+L24	Sustained + Accelerations + Relief Valve Reaction	(OCC)	Scalar	Disp/Force/Stress
27	L25+L23	Sustained + Acceleration + Wind	(OCC)	Scalar	Disp/Force/Stress
28	L26+L27	Maximum Stress (Sustained + Occasional)	(OCC)	MAX	Disp/Force/Stress
29	L9+L10+L11	Max Support Loads (Design)	(OPE)	MAX	Force
30	L3+L22	Operating Loads (Rotating Equipment)+Acc.	(OPE)	Scalar	Disp/Force
31	L1+L6+L7+L8	Design Loads (Equipment)	(OPE)	MAX	Force

Load	*Description*	*Caesar II Load Identifier*
T1	(Thermal 1) Thermal expansion from maximum temperature above ambient conditions	Temp 1
T2	(Thermal 2) Thermal expansion from minimum temperature below ambient conditions	Temp 2
T3	(Thermal 3) Thermal expansion from normal operating conditions	Temp 3
U1	(Accel 1) Acceleration from wave motion (Pitch)	Uniform G Load Vector 1
U2	(Accel 2) Acceleration from wave motion (Heave)	Uniform G Load Vector 2
U3	(Accel 3) Acceleration from wave motion (Roll)	Uniform G Load Vector 3
W	(Weight) Normal operating weight with contents	Dead Weight with Contents
P1	(Pressure) Design Pressure	Pressure 1
H	(Force) Spring hanger loads	Spring force
F1	(Force) Relief valve reactions	Force Vector 1
WIN	(Wind 1) Maximum Wind in the -X-direction	Wind Load #1
WIN	(Wind 2) Maximum Wind in the +Z-direction	Wind Load #2
WW	Weight with water content	Weight of Water

APPENDIX H
SUBSEA LOAD CASE MATRIX

A typical load case matrix for a subsea manifold is shown on the next page. Refer also section 4.4.2 for other calculations to be performed.

Load description:

L1 Pressure test at site (on shore)
L2 Lifting analysis in-shore (accelerations, deformations)
L3 Barge transportation analysis, barge motions
L4 Barge transportation analysis, green sea impact loads
L5 Lowering of installation through splash zone (wave slamming)
L6 Landing on seabed (g-forces)
L7 Pipeline Tie-in analysis
L8 Pressure test offshore, subsea commissioning
L9 Max operating or design condition
L10 Occasional and accidental conditions

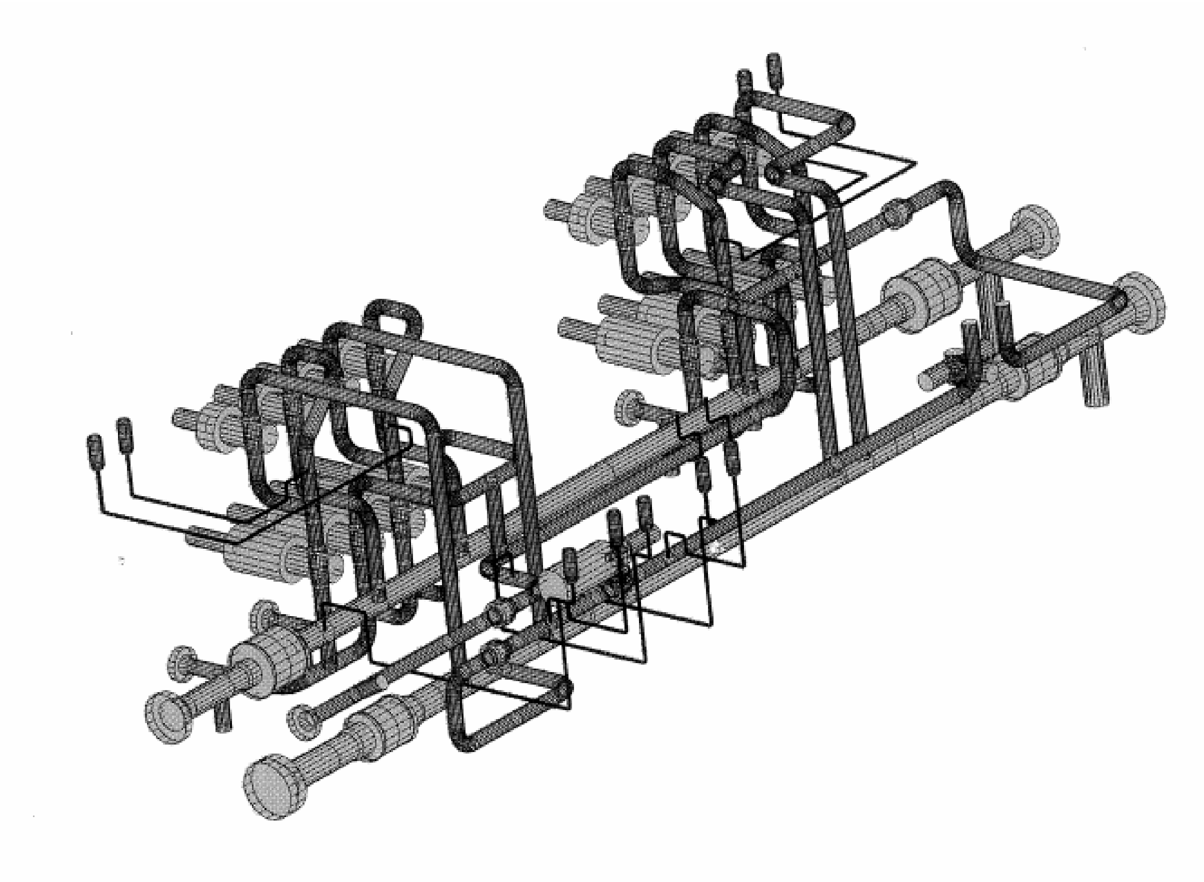
Relevant combinations of L1 to L10 are to be performed based on the analyst judgement. It is not required to combine two occasional, two accidental, or a combination of one occasional and one accidental load in the same load case.

The load case matrix is presented on the next page.

Parameters	*Load cases*									
	L1	*L2*	*L3*	*L4*	*L5*	*L6*	*L7*	*L8*	*L9*	*L10*
Weight of piping and insulation	X	X	X	X	X	X	X	X	X	X
Test pressure, at site/onshore	X									
Test pressure offshore								X		
Design pressure									X	X
Design temperature (high/low)									X	X
Ambient temperature air	X	X	X	X						
Ambient temperature sea bottom					(X)	(X)	X	X	X	X
Weight of internal fluid used for pressure test	X							X		
Weight of internal fluid during operation									X	X
Buoyancy					(X)	X	X	X	X	X
Accelerations, lifting		X			X					
Wind loads		X	X	X	X					
Barge, wave accelerations			X							
Green sea impact load during transportation				X						
Wave slamming loads, splash zone					X					
Seabed landing, impact/retardation						X				
Sea bottom current									X	X
Slugging									X	
Fluid hammer, and surge loads									X	(X)
Tie-in loads							X			
External operational loads									X	X
Friction							X		X	X
Settlement							(X)	(X)	X	X
Trowel board and ROV impact loads, Earthquake etc										X

APPENDIX I
SUBSEA PIPE STRESS MODEL

A typical pipe stress model of a subsea production manifold with valves is shown below. All pipe-supports have been removed from the plot for better visibility.



APPENDIX J
FATIGUE CALCULATION EXAMPLE

The next two pages show an example on how fatigue calculations can be performed according to PD5500.
The purpose is only to show a format based on PD5500, Annex C, and PD5500 working example W.6.2.3.
Refer section 3.12.4 in this RP for a description of the piping being analysed.

13.02.2008 16:00

Project	Example calculation. The purpose is only to show a typical format and methodology for fatigue calculations according to PD5500.
Line No.	Typical bridge-piping fatigue calculation. Stresses are calculated from pipe stress analysis at a pipe bend in a bridge landing area on one of the two platforms.
Design temp	
Operating temp	
Operating pressure	
Test pressure	
Pipe material	
User	

INPUT:

Fatigue Class (C, D, E, F, F2, G, W)	F2	- (Ref. PD 5500 Annex C, Table C1)
S-N constant m for $N < 10^7$ cycles	3	- (Ref. PD 5500 Annex C, Table C1)
S-N constant A for $N < 10^7$ cycles	4,31E+11	- (Ref. PD 5500 Annex C, Table C1)
S-N constant m for $N > 10^7$ cycles	5	- (Ref. PD 5500 Annex C, Table C1)
S-N constant A for $N > 10^7$ cycles	5,25E+14	- (Ref. PD 5500 Annex C, Table C1)
Counting of cycles are number of cycles during a design life of	30	Years
E - module of pipe material at operating temperature	200000	MPa
Nominal wall thickness of pipe material	12,7	mm

Fatigue calculation according to PD 5500 2003 Edition
(Ref. Annex C and Working Example W.6.2.3 Simplified Fatigue Analysis)
Programmed by Jan H. Hansen 2005

Individual loading events for bridge piping						Combined loading events	
Source	Description	Bridge movement dX	Stress Range	(Mpa)	Estimated No. of Cycles ni during designlife time	Assumed possible combinations Sri	MPa
A1	Platform setlement (+400 mm)	+ 400 mm	SA1	100	1	Sr1=SA1+SA2+SA14+SA24+SA31	542
A2	Wave 24.3 m (100 year wave)	+/- 500 mm	SA2	235	1	Sr2=SA2+SA14+SA24+SA31	442
A3	Wave 23 m	+/- 455 mm	SA3	214	1	Sr3=SA3+SA14+SA24+SA31	421
A4	Wave 22m	+/- 422 mm	SA4	198	1	Sr4=SA4+SA14+SA24+SA31	405
A5	Pressure testing (eg. 1 cycle each 10. year)	NA	SA5	40	3	Sr5=SA5+SA27	190
A6	Wave 21 m	+/- 390 mm	SA6	183	4	Sr6=SA6+SA14+SA24+SA31	376
A7	Wave 20 m	+/- 359 mm	SA7	169	16	Sr7=SA7+SA14+SA24+SA31	247
A8	Wave 19 m	+/- 329 mm	SA8	154	69	Sr8=SA8+SA14+SA24+SA31	361
A9	Wave 18 m	+/- 300 mm	SA9	141	139	Sr9=SA9+SA14+SA24+SA31	348
A10	Wave 17 m	+/- 272 mm	SA10	128	208	Sr10=SA10+SA14+SA24+SA31	335
A11	Wave 16 m	+/- 246 mm	SA11	115	371	Sr11=SA11+SA14+SA24+SA31	322
A12	Wave 15 m	+/- 220 mm	SA12	103	556	Sr12=SA12+SA14+SA24+SA31	310
A13	Wave 14 m	+/- 195 mm	SA13	92	1204	Sr13=SA13+SA14+SA24+SA31	299
A14	Max to Min design temp variation (eg. 1 cycle each week)	NA	SA14	193	1560	Sr14=SA14+SA24+SA31	207
A15	Wave 13 m	+/- 173 mm	SA15	81	1807	Sr15=SA15+SA19+SA24+SA31	115
A16	Wave 12 m	+/- 151 mm	SA16	70	3613	Sr16=SA16+SA19+SA24+SA31	104
A17	Wave 11 m	+/- 130 mm	SA17	61	10801	Sr17=SA17+SA19+SA24+SA31	95
A18	Wave 10 m	+/- 111 mm	SA18	52	22931	Sr18=SA18+SA19+SA24+SA31	86
A19	Temp. variation +/- 20% of operating (eg. 4 cycles each day	NA	SA19	20	43800	Sr19=SA19+SA24+SA31	34
A20	Wave 9 m	+/- 92 mm	SA20	43	49164	Sr20=SA20+SA24+SA31	57
A21	Wave 8 m	+/- 76 mm	SA21	35	120630	Sr21=SA21+SA24+SA31	49
A22	Wave 7 m	+/- 60 mm	SA22	28	253768	Sr22=SA22+SA24+SA31	42
A23	Wave 6 m	+/- 46 mm	SA23	21	611210	Sr23=SA23+SA24+SA31	35
A24	Pressure fluctation +/- 10% (eg. 100 each day)	NA	SA24	10	1095000	Sr24=SA24+SA31	14
A25	Wave 5 m	+/- 34 mm	SA25	16	1520887	Sr25=SA25+SA31	20
A26	Wave 4 m	+/- 23 mm	SA26	11	3960499	Sr26=SA26+SA31	15
A27	Wave 3 m	+/- 14 mm	SA27	7	10642398	Sr27=SA27+SA31	11
A28	Wave 2 m	+/- 7 mm	SA28	3	30039364	Sr28=SA28+SA31	7
A29	Wave 1 m	+/- 2 mm	SA29	1	88467629	Sr29=SA29+SA31	5
A30	Wave 0.25 m	+/- 0 mm	SA30	1	55645993	Sr30=SA30+SA31	5
A31	Slugs matching natural frequency (eg. 5 Hz) over 30 years	NA	SA31	4	4730400000	Sr31=SA31	4

Page 1 of 2

Page 2 of 2 (Fatigue calculation of bridge piping started at the previous page).

Fatigue calculation for combined loading events

Combined Loading	Stress range for combined loading events Mpa	m	A	No. of Cycles ni	Ni	ni/Ni
Sr1	542	3	4,31E+11	1	2372	0,000422
Sr2	442	3	4,31E+11	1	4374	0,000229
Sr3	421	3	4,31E+11	1	5062	0,000198
Sr4	405	3	4,31E+11	1	5685	0,000176
Sr5	190	3	4,31E+11	3	55064	0,000054
Sr6	376	3	4,31E+11	4	7105	0,000563
Sr7	247	3	4,31E+11	16	25063	0,000638
Sr8	361	3	4,31E+11	69	8028	0,008595
Sr9	348	3	4,31E+11	139	8962	0,015510
Sr10	335	3	4,31E+11	208	10046	0,020705
Sr11	322	3	4,31E+11	371	11313	0,032795
Sr12	310	3	4,31E+11	556	12678	0,043856
Sr13	299	3	4,31E+11	1204	14129	0,085214
Sr14	207	3	4,31E+11	1560	42581	0,036636
Sr15	115	3	4,31E+11	1807	248333	0,007277
Sr16	104	3	4,31E+11	3613	335760	0,010761
Sr17	95	3	4,31E+11	10801	440512	0,024519
Sr18	86	3	4,31E+11	22931	593790	0,038618
Sr19	34	5	5,25E+14	43800	9272210	0,004724
Sr20	57	3	4,31E+11	49164	2039407	0,024107
Sr21	49	3	4,31E+11	120630	3210260	0,037576
Sr22	42	3	4,31E+11	253768	5097773	0,049780
Sr23	35	5	5,25E+14	611210	8021168	0,076200
Sr24	14	5	5,25E+14	1095000	783317218	0,001398
Sr25	20	5	5,25E+14	1520887	131652125	0,011552
Sr26	15	5	5,25E+14	3960499	554780970	0,007139
Sr27	11	5	5,25E+14	10642398	2615859569	0,004068
Sr28	7	5	5,25E+14	30039364	25066150974	0,001198
Sr29	5	5	5,25E+14	88467629	134811775812	0,000656
Sr30	5	5	5,25E+14	55645993	134811775812	0,000413
Sr31	4	5	5,25E+14	4730400000	411412890052	0,011498
					Σni/Ni =	0,54
					Estimated fatigue life L=	55 Years

CONCLUSION:

This simplified fatigue analysis is valid if:

$$\boxed{\Sigma \frac{n_i}{N_i} \leq 0.6\left(\frac{22}{e}\right)^{0.75}}$$

e= 22

Σni/Ni = 0,54

$0.6*(22/e)^{0.75}$= 0,60

Ref. PD5500 Annex C , Equation (C.4)
"e" is the greatest value of 22mm or pipe nominal wall thickness!

Conclusion: Calculation methodology is valid!

NOTE!
The methodology above is not valid for fatigue of risers. Fatige calculations of risers and pipelines requires the use of a Design Fatigue Factor, DFF, in the range 3-10. Ref. API RP 2RD or DNV-RP-F204.